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RELATION OF SCAVENGING RATIO AND SCAVENGING EFFICIENCY IN THE TWO-STROKE COMPRESSION IGNITION ENGINE

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RELATION OF SCAVENGING RATIO AND SCAVENGING EFFICIENCY IN THE TWO-STROKE COMPRESSION IGNITION ENGINE

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1950

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18 May 1950

Professor Joseph S. Newell
Secretary of the Faculty
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Dear Sir:

In partial fulfilment of the requirement for the degree of Naval Engineer, from the Massachusetts Institute of Technology, we hereby submit our thesis entitled: RELATION OF SCAVENGING RATIO AND SCAVENGING EFFICIENCY IN THE TWO-STROKE COMPRESSION IGNITION ENGINE.

Respectfully,

ACKNOWLEDGMENT

The authors wish to express their appreciation to Professors C. F. Taylor, A. R. Rogowski, W. A. Leary, and P. M. Ku for their supervision and constructive criticism.

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SUMMARY

The indicated horsepower of a two-stroke diesel engine varies directly with scavenging efficiency, other things being constant. Therefore, any improvement in scavenging efficiency (e_s) at constant scavenging ratio (R_s) will increase the indicated horsepower. The object of this thesis has been to determine the curves of e_s versus R_s for three engines using different systems of scavenging. From these curves, the relative merits of the three systems have been determined.

The method used was to draw curves of fuel air ratio versus exhaust gas components based on exhaust samples taken from two four-stroke engines where the fuel air ratio was known. The samples were taken with a sampling valve placed in the exhaust manifold. The samples were analyzed for exhaust components using an Orsat gas analyzer. With the curves developed on the four-stroke engines as a basis, the fuel air ratio of the three two-stroke engines was determined. This was done by measuring the fuel supplied, air supplied, and determining the fuel air ratio by exhaust gas samples. From this data, curves of e_s versus R_s were drawn up for the three engines.

Results

The curves of fuel air ratio versus exhaust gas components (Figure 7) obtained on the four stroke engine agreed with the results of other experiments. The curves of e_s versus R_s for the loop scavenged engine (Figure 10) are in agreement with previously published curves of the MIT 2 stroke loop scavenged engine.

The curves of e_s versus R_s for the through scavenged ported engine (Figure 8) and poppet valve engine (Figure 9) are reasonable. No earlier data is available for comparison.

Of the three engines tested, the General Motors 3-71, poppet valved, through scavenged engine showed the highest e_s for any particular value of R_s. The Fairbanks Morse model 38E5¼ through scavenged opposed piston engine, was next highest. Both these engines showed better values of e_s than those for complete mixing. (See Figure 11).

The M.I.T. modified C.F.R. loop scavenged engine showed the lowest values of e_s for any particular value of R_s. This engine evidenced mixing and short circuiting of scavenging air.

The values of scavenging mean effective pressure for the engines were in exactly the same relation as the values for e_s, the General Motors engine being the highest. This would indicate that although the e_s is highest in the G.M. engine, it does not mean a higher brake mean effective pressure. Carrying the scavenging ratio beyond 1.0 has increasingly diminishing returns for the scavenging pump power expended. Emphasis should be placed on port flow coefficients instead of wasting power in driving a scavenging pump to achieve a high e_s.

The field should be further investigated and these results improved by examination of isolated cylinders. Although piston speed is a weak function in the relation of e_s to R_s, the effect should be investigated. The effect of fuel-air ratio on the e_s, R_s relation should be established.

INTRODUCTION

The increasing number of compression ignition engines designed to operate on the two-stroke cycle principle demands furthur investigation of this method of operation. The improvement in output of two-stroke engines usually centers around the problem of obtaining good scavenging efficiencies as shown by the formula for IHP: $IHP = \rho_s e_s N V_c F E_c \eta_I$. For a given bore, stroke, and RPM, $IHP = K \rho_s e_s \eta_I$. Any improvement in e_s, at a given R_s, will increase engine output. Yet, the relationship of e_s to R_s is one of the few items of the internal combustion engine that has not been investigated to the degree necessary for sound design decisions.

The difficulty in the past has been the lack of a simple, accurate, and easy-to-use device for obtaining a sample of the combustion products of the engine before they have been diluted by the scavenging air.

The purpose of this thesis was to develop such a device, as outlined above, and to then investigate e_s versus R_s for the three types of scavenging generally used for two-stroke diesel engines.

A simple type of sampling check valve placed in the exhaust manifold had been suggested to a previous thesis group, (MEASUREMENT OF SCAVENGING EFFICIENCY OF THE TWO-STROKE ENGINE: A COMPARISON AND ANALYSIS OF METHODS) by Professors C. F. Taylor and A. R. Rogowski . Some preliminary tests were made with this sampling valve, but no definite conclusions were reached as to its use.

As a starting point for this thesis, it was decided to further investigate the use of this valve.

In the past attempts to gather samples of the cylinder combustion products from the cylinder required complicated, mechanically-timed sampling valves, since the valve must be closed during the scavenging period. The basic idea of this valve is that if it can be placed close enough to the exhaust ports or valve to utilize the dynamic pressure of the exhaust gases during the cylinder blow-down period, a sufficiently high sampling pressure can be built up in a receiver behind the check valve, to keep it closed against scavenging pressure.

The valve was first tested using a standard CFR four-stroke, spark ignition engine to see if a representative sample of the exhaust gases could be gathered, and if a sufficiently high sampling pressure could be built up. The engine was run over a wide range of fuel air ratios and the exhaust gas samples were analyzed for CO_2, CO, and O_2 using an Orsat Analyzer.

The samples analyzed checked well with the published curves of exhaust gas content versus fuel air ratio by De Alleva and Lovell. The conclusion reached was that a representative sample of the exhaust gases could be obtained using a sampling valve of this type.

Pressure well in excess of any scavenging pressures used for two-stroke engines was maintained in the receiver. Therefore, it was concluded the check valve would remain closed dur-

ing the scavenging period.

There were no published curves of exhaust gas content versus fuel-air ratios in the range of diesel engine operation (F/A .02 - .05). The next step was to develop these curves. A four-stroke CFR engine, modified by use of a Comet head precombustion chamber, and equipped with a unit fuel injection valve, was used for this part of the investigation. The samples were analyzed for CO_2 and O_2 (no CO is present at low fuel air ratios) and smooth curves were developed (see Figure 7). It was again possible to build up satisfactory back pressures in the exhaust sample receiver.

The three types of two-stroke engines investigated for the development of scavenging efficiency (e_s) versus scavenging ratio (R_s) curves were:

1. A three cylinder, two-stroke opposed piston Fairbanks Morse Engine Model 38E5¼ representing the "through scavenging" type with opposed pistons.
2. A 3-71, GM, two-stroke diesel model representing the "through scavenging" type with poppet valves in the head.
3. A single cylinder, two-stroke, gasoline engine equipped with unit fuel injection representing the "loop scavenging" type.

The products of combustion were gathered during the exhaust blow down period with a sufficiently high sampling pressure built up in the receiver to insure that the sampling check valve

was closed during the scavenging process. The exhaust samples were analyzed, scavenging efficiency and scavenging ratio calculated, and the points plotted (see Figures 8, 9, 10, 11).

In the multi-cylinder engines, exhaust samples were analyzed from each cylinder but no attempt was made to isolate individual cylinders - due primarily to lack of time. Since the calculations of e_s and R_s are based on the total amount of air and fuel used, the assumption was made that air and fuel were evenly divided between the cylinders. In each case the result of individual cylinder analysis is plotted and the curve of e_s versus R_s is drawn through the mean of the points.

PROCEDURE

The procedure outlines the steps taken to obtain the data and results presented in this paper. The many difficulties, the blind alleys, and the lessons learned are included in the appendix.

The procedure falls into five separate categories:

1. Determination of the relation of fuel-air ratio to some of the exhaust gas components (CO_2, Co and O_2), in the four-stroke, spark ignition engine by the use of a sampling check valve in the exhaust blow-down stream.
2. Determination of the relation of fuel-air ratio to some of the exhaust gas components (CO_2, CO, O_2), in the four-stroke, compression ignition engine by the use of a sampling check valve in the exhaust blow-down stream.
3. Determination of the relation of scavenging ratio to scavenging efficiency in a two-stroke, opposed piston, compression ignition engine at a single piston speed.
4. Determination of the relation of scavenging ratio to scavenging efficiency in a two-stroke, poppet valve, compression ignition engine at a single piston speed.
5. Determination of the relation of scavenging ratio

to scavenging efficiency in a two-stroke, loop scavenged, gasoline injection, spark ignition engine.

FOUR-STROKE SPARK IGNITION ENGINE

A C.F.R. engine was used for this part of the investigation. A sampling check valve* was inserted into the exhaust pipe. The position of the valve tip was adjusted to receive the full impact of the blow-down pressure wave. The valve was connected to the sampling apparatus as shown in the schematic diagram.** It was possible to get a pressure behind the sampling valve (sampling pressure) of 17" Hg. when the valve was correctly adjusted. The sampling pressure was never allowed to drop below 12" Hg., while sampling, to develop a technique for future experiments with two-stroke engines. An Orsat*** apparatus was used to analyze the gas samples.

After the engine had been run to the point of thermal equilibrium, samples were taken of the exhaust gas at various fuel-air ratios. Each sample taken was analyzed and plotted. All points appear in the finished plot. All runs were at full throttle. Carburetion was accomplished by using a steam jacketed vaporizing tank. The speed was kept constant by the use of an electric dynamometer. Air consumption was measured by the

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* Plate 4 and appendix.

** Plate 2 (the arrangement of the G.M. and the C.F.R. are the same).

*** See appendix.

pressure drop across a square edged orifice.* Fuel consumption was measured by a rotometer. The fuel was 100 octane gasoline. The fuel-air ratio was varied from .076 to .05. Care was taken to completely purge the exhaust system between changes of fuel-air ratios.

FOUR-STROKE COMPRESSION IGNITION ENGINE

A C.F.R. engine, modified by use of a Comet head (precombustion chamber), was used for this part of the investigation. The sampling valve was inserted into the exhaust pipe proximate to the exhaust valve to receive the full impact of the exhaust blow-down. The procedure used in this phase of the investigation was similar to that used in the previous step. The fuel-air ratio was varied in this case from .02 to .06. The relation between fuel-air ratios and exhaust gas components from .02 to .076 was established as a result of steps one and two.**

THE FAIRBANKS MORSE MODEL 38, E5¼ TWO-STROKE, OPPOSED PISTON, COMPRESSION IGNITION ENGINE

The engine and related equipment was set up as described in the schematic diagram.*** Full load was applied to the engine through its direct coupled generator. Electrical power was absorbed in a resistance grid. The engine was run at

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* Figure 1 and details of air measurement in the appendix.

** Figure 7.

*** Plate II

1200 r.p.m. (1,450 ft./min. piston speed). The air inlet to the scavenging blower was wide open. Air and fuel consumption were measured as in steps one and two.

After thermal equilibrium was reached, sampling was begun. The sampling check valve was inserted through the pyrometer inlet. The valve was adjusted to receive the full impact of the blow-down exhaust gas. A sampling pressure of 10" Hg. maximum was obtained. The scavenging air pressure was 6" Hg. Sampling of the exhaust gas was done with a sampling pressure of 7" Hg.

A series of five separate analyses were taken from each of the three cylinders. Two samples were analyzed with the Orsat equipment, and three with the Hays flue gas analyzer. It was necessary to shut the engine down each time the sampling valve was shifted from one cylinder to another, but in each case the original operating conditions were reproduced.

In order to find the relation of e_s to R_s at a different point, the air inlet to the scavenging air blower was throttled. The load was then adjusted downward, to produce the same fuel-air ratio as was obtained at full load and wide open air inlet. To obtain the value of the load to give the correct fuel-air ratio, the number one cylinder exhaust was analyzed until the gas components checked with the previous run. The exhaust of all three cylinders was then analyzed as was done in the previous run. The method of translating the exhaust sample analyses into the scavenging ratio and scavenging efficiency for

this and all other engines is included in the sample computations to be found in the appendix.

THE GENERAL MOTORS SERIES 3-71 POPPET VALVE, COMPRESSION IGNITION ENGINE

The 71 series engine with its related equipment is shown in the schematic diagram appended.* This engine had a single water cooled manifold to handle the exhaust gas from all cylinders. This manifold was pierced opposite each exhaust passage and fittings brazed in place to accommodate sampling valves. Each cylinder had two exhaust valves. Only one was utilized to sample the exhaust gas.

Air and fuel consumption were measured as before. Sampling of the exhaust was started after thermal equilibrium was reached. A sampling pressure of 14" Hg. maximum was obtained after the sampling valves were correctly adjusted. Sampling was done with a sampling pressure of 12" Hg. The scavenging air pressure at full load was 11" Hg.

As in the case of the FM engine, the first samples were taken with the engine fully loaded. The air inlet to the scavenging blower was wide open. The load was applied through a resistance loaded electric dynamometer. The engine speed was held constant at 1740 r.p.m., (1,450 ft./min. piston speed). All three cylinders were checked in succession and then rechecked to insure reproducibility.

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* Plate #I

The e_s and R_s were changed in the same manner as with the FM engine and again all cylinders were checked twice. At least five Orsat readings were taken during each check of a single cylinder.

The interpretation of the Orsat readings and their translation into e_s and R_s values is to be found in the appendix as part of the sample calculations.

LOOP SCAVENGED, M.I.T. MODIFIED C.F.R. ENGINE WITH GASOLINE INJECTION AND SPARK IGNITION

The engine and its related equipment was set up as described in the schematic diagram.* Air and fuel consumption were measured as before. Scavenging air was supplied from an outside air compressor which did not draw on the engine for its motive power. Unit gasoline injection was accomplished with all ports closed. The speed was maintained constant at 1,930 r.p.m. (1,450 ft./min. piston speed). The load was applied through a resistance loaded electric dynamometer. The sampling valve was placed in the exhaust port about 1/16" from the inside cylinder wall and at an angle of 60° to the axis of the cylinder.

Scavenging pressure at full load was 8" Hg. The sampling pressure was 30" maximum. The sampling pressure was maintained at 9" during the sampling process. The cylinder was checked at full load. Two more sets of analyses were made at different values of scavenging ratio and scavenging efficiency. These

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* Plate III.

points were obtained as before, by reducing the scavenging air pressure and the load, but maintaining the fuel-air ratio and speed a constant. No difficulty was experienced in obtaining sufficient back pressure at any point to insure obtaining a true sample of exhaust gas. The interpretation of the results of the exhaust gas analyses and translation into terms of e_s and R_s are to be found in the sample calculation.

RESULTS

1. The relation between fuel-air ratio and three of the components of exhaust gas, (CO_2, CO and O_2) was established over a range of fuel-air ratios from .02 to .09. (See Figure 7.)

2. A sampling check valve, originally suggested by Professors C. F. Taylor and A. R. Rogowski, was modified and used successfully to collect representative samples of the exhaust gas in the two-stroke internal combustion engine.

3. The relation of scavenging efficiency to scavenging ratio in the M.I.T. modified C.F.R. engine, the General Motors 3-71 engine, and the Fairbanks Morse model 38E5¼ engine was established at a piston speed of 1,450 ft./min. (See Figure 11).

4. The relation of the scavenging efficiency to scavenging ratio in the M.I.T. modified C.F.R., gasoline injection, two-stroke, loop scavenged, engine as determined by use of the sampling check valve, compared favorably with the same relation previously computed for the M.I.T. two-stroke engine.

DISCUSSION OF THE RESULTS

FUEL-AIR RATIO VS EXHAUST GAS COMPONENTS

At the start of the investigation there were available the relations previously established by D'Alleva and Lovell (1) and Gerrish and Means (2). After this phase of the investigation was completed the relations established by Elliot and Davis (3) were made available. The latter covered the compression ignition type engine; the former two, the spark ignition engine. All engines were four-stroke.

The results of this investigation were plotted not as a single line but rather as a band to include the variations in the analyses. No attempt was made to rationalize the results by application of an arithmetical average or by the method of least squares. It was reasoned that successive analysis of the two-stroke scavenged engines would contain the same variations as in the four-stroke investigations and that any attempt to reduce the scatter to a single line would introduce errors in the future use of the curves.

The results of the previous independent investigations all fall within the band produced by this investigation.

Beyond the chemically correct fuel-air ratio (.0665) there is a greater scatter in the analyses than in the leaner regions. This was found to be true in the references used for comparison as well as in the results of this investigation.

The reason is not immediately apparent. With the use of a steam jacketed vaporizing tank instead of the usual means of carburization, it was expected that a uniform, homogeneous mixture would be fed to the cylinder. This would seem to exclude the possibilities of stratification and poor mixing effects.

It might be reasoned that the increase in fuel-air ratio would lead to higher cylinder pressures and temperatures between the chemically correct mixture and the best power mixture, and therefore the physical conditions of combustion of the charge would be altered. This is undoubtedly true, but with the engine running at thermal equilibrium and the variants held constant, it does not explain the variation in the exhaust gas analysis at the same fuel-air ratio.

OPERATION OF THE SAMPLING CHECK VALVE

The valve was placed in the exhaust piping so that it received the earliest part of the exhaust gas blow-down. The pressure maintained behind the check valve was never allowed to go below scavenging air pressure. It was usually maintained at from 1/2 to 1 inch of mercury above the scavenging air pressure to insure no contamination of the sample.

All analyses of exhaust gas samples taken with the sampling pressure above but close to the scavenging air pressure were consistent and reproducible. When the scavenging air pressure was higher than the sampling pressure, the analyses were erratic

and could not be reproduced. The O_2 content of such samples increased noticeably.

It might be argued that the sampling valve extracts its sample at a particular point in the blow-down only and thus might conceivably be affected by stratification in the cylinder. This is not true. It is true that stratification in the cylinder does exist, particularly at the lower fuel-air ratios. If the check valve were mechanically operated to extract a sample during a short interval during blow-down, then it is conceivable that the same strata might be sampled each cycle. The same condition might exist if the spring behind the check valve were to exert enough force to cause it to operate only over a small range of pressures.

The gaseous loaded check valve eliminates all these possibilities by providing a variable "spring". It can thus extract an exhaust gas sample over any range of pressures from the maximum blow-down pressure on down. The valve can be made to do this because it has a very narrow seat. The line of contact between the seat and the valve was made by lapping two cones of different angles together. The line of contact is narrow enough so that the area exposed to exhaust blow-down can be considered equal to that exposed to the sampling pressure on the valve. At the start of sampling the spring load is such that it can be overcome by the scavenging pressure alone. The valve therefore passes any and all products coming through the exhaust passage. If the sampling pressure so formed is allowed

to build up behind the valve, and the leakage past the valve is negligible, then soon the sampling pressure acting on the valve will approximate the maximum blow-down pressure. At this point extraction will cease. The sampling pressure on the valve can be regulated by bleeding off the excess to the atmosphere. If the sampling pressure is held just below that of the maximum blow-down pressure, the valve will be very selective and admit only the products of the first part of the blow-down. By reducing the sampling pressure to a point just above that of the scavenging air pressure, the valve can be made to extract a sample of the exhaust from the start of blow-down to the beginning of scavenging. The sampling pressure was held one inch of mercury above the scavenging pressure to insure no contamination due to dynamic effects.

After prolonged operation in the gasoline two-stroke engine without cooling the valve tip, the spring was found to have been permanently deformed to the point where it could not possibly bear against the valve. This was adequate proof that the spring lost its significance after the pressure was once built up. It is not too far from fact to hazard a guess that the spring could be left out of the valve entirely. The sampling pressure could be built up with compressed gas before operation and the valve would operate normally.

SCAVENGING EFFICIENCY VS SCAVENGING RATIO IN THE M.I.T. MODIFIED C.F.R. ENGINE

Figure 10 is a plot of the results obtained from the runs on the M.I.T. C.F.R. Engine. This was a single cylinder engine. The geometry of the exhaust system made it possible to locate the sampling valve in the full blast of the initial exhaust blow-down. For this reason, sampling pressures as high as 30" Hg. were obtained. During sampling, the sampling pressure was maintained 1" Hg. above scavenging pressure.

The samples taken were consistent over the whole series of runs. The values of e_s versus R_s are plotted in Figure 10 along with the curve of e_s versus R_s for the M.I.T. loop scavenged, two-stroke engine. The M.I.T. two-stroke engine is geometrically similar to the M.I.T. Mod. C.F.R. Engine. The agreement of the results of this thesis with the curves of the M.I.T. two-stroke obtained by an entirely different method seem to verify the assumption that it is possible to obtain accurate results using the sampling valve technique.

SCAVENGING EFFICIENCY VS SCAVENGING RATIO IN THE GENERAL MOTORS 3-71 ENGINE

Figure 9 is a plot of the results obtained from the runs on the GM 3-71. This engine had two exhaust valves per cylinder. The exhaust manifold arrangement made it very difficult to place the sampling valve in the proper position to obtain high sampling pressures. The initial exhaust samples were taken with a sampling pressure of 9" Hg. This was 2" Hg. below the scavenging pressure. It was reasoned that the pressure drop through the inlet and exhaust ports would be greater than 2" Hg. and that therefore the sampling check valve would remain closed during the scavenging process. When samples were taken on this basis and the results plotted on the e_s versus R_s curve the points were scattered. The majority of points gave a high value of e_s. This indicated a lean mixture or contamination of the sample with scavenging air (i.e. the valve was not remaining closed during the scavenging process).

After the position of the sampling valve was changed to maintain a sampling pressure above scavenging pressure at all times, a new series of runs were made. Under these conditions, the points of all three cylinders were consistent and reproducible. As a result, it was reasoned that, due to dynamic effects during the scavenging process, a pressure of 9" Hg. was not high enough to prevent contamination of the samples with scavenging air. For this reason, the initial runs at a sampling pressure of 9" Hg. are omitted from the composite plot. (Figure 11).

SCAVENGING EFFICIENCY VS SCAVENGING RATIO IN THE FAIRBANKS MORSE 38E5¼

Figure 8 is a plot of the results obtained from the results on the FM 38E5¼ engine. The Fairbanks Morse three-cylinder engine was the first two-stroke engine to be tested. Total air and fuel flow to the three cylinders was measured, and it was assumed that the air and fuel distributed equally among the three cylinders. The first run was made at 1,242 r.p.m. (1,500 ft./min. piston speed), which was 42 r.p.m. above rated speed. The maximum load possible was applied. A series of points at different values of R_s were taken from exhaust samples from a single cylinder. For the next series of runs, the engine speed was reduced to 872 r.p.m. When the results of the 872 r.p.m. runs were plotted on an e_s versus R_s curve (Figure 8) the points were very erratic. It was reasoned that these erratic results were caused by poor distribution of air or fuel between the cylinders, when the engine operated below rated speed. A check of samples from all three cylinders at 872 r.p.m. and constant R_s showed a wide variation in e_s. Samples were then taken from all three cylinders at rated speed. These samples showed little or no variation in e_s at constant R_s. This seemed to verify the reasoning of unbalance in fuel or air distribution at other than rated speed ~~load~~. It was very difficult in this engine to measure the flow of air and fuel to a single cylinder. For these reasons the runs at 872 r.p.m.

were omitted from the composite curves, Figure 11. The runs at 1,242 r.p.m. seemed to check well with the points for the FM engine plotted on Figure 11. These points were omitted from Figure 11 because they were at a different piston speed.

COMPOSITE PLOT OF SCAVENGING EFFICIENCY VS SCAVENGING RATIO FOR THE THREE REPRESENTATIVE ENGINES

The composite plot shows that at any particular value of R_s, the value of e_s is highest for the G.M. series 71 engine. The F.M. engine is next highest.

The G.M. has a value of e_s for any particular value of R_s which is at all times higher than that for the complete mixing curve.

This is also true of the F.M. engine at values of R_s less than 1.2. Above the R_s value of 1.2 the F.M. shows a lesser value of e_s than complete mixing, indicating possible short circuiting effect of the scavenging air.

The M.I.T. modified C.F.R. engine, which is loop scavenged, shows both a condition of some mixing and some short circuiting exists. This was to be expected.

The comparison of the F.M. and G.M. engines is conducive of more speculation. While recording the data for this investigation, data on the blower was also taken to aid in reproducing operating conditions.

Using this data an attempt was made to determine the flow coefficients of the engines by utilizing the orifice equation.

$$\dot{M}_a = C A \rho_i C_{si} \phi_2$$

and

$$\frac{CA}{A_p} = \frac{\dot{M}_a}{\rho_{si} C_{si} \phi_2 A_p}$$

where:

$\dot{M}_a$ = flow rate

ρ_i = inlet density

C_{si} = speed of sound in air

ϕ_2 = compressibility coefficient

A_p = area piston

A = port area

With a knowledge of the port coefficients, a set of standard operating conditions was assumed for comparison purposes. R_s was then computed from the relation:

$$R_s = 2\,\frac{CA}{A_p}\left(\frac{r-1}{r}\right)\left(\frac{\rho_i}{\rho_e}\right)\left(\frac{C_{si}}{S}\right)\phi_2 \quad *$$

Still utilizing the assumed conditions scavenging mean effective pressure (SMEP) was computed:

$$SME = J R_s \rho_s c_p \left(\frac{r}{r-1}\right) T_2 \left[\left(\frac{P_s}{P_2}\right)^{.285} - 1\right] \div \eta_p$$

This comparison showed that the G.M. series expended more power in scavenging than did the F.M. Thus it is possible that the brake mean effective pressure of an engine may be lowered instead of raised by trying to achieve a high scavenging efficiency.

The authors cannot state that this is specifically the case in this instance, because the data on the blowers used to make this analysis is not considered sufficient to drawing a definite conclusion.

$$* \quad \phi_2 = \sqrt{\frac{2}{K-1}\left[\left(\frac{P_s}{P_2}\right)^{\frac{2}{K}} - \left(\frac{P_s}{P_2}\right)^{\frac{K-1}{K}}\right]} \qquad K = 1.4$$

RECOMMENDATIONS

The results obtained by use of a sampling valve to analyze exhaust gases on a two-stroke engine prove the feasibility of this method of investigation. The following recommendations are made for future investigation based on the results of this thesis:

1. Investigation of the relation of e_s and R_s at piston speeds other than rated, after isolation of cylinders has been completed.

2. Investigation of the relation of fuel-air ratio and e_s, holding R_s and speed a constant.

3. Investigation of the relation of e_s and R_s on one particular engine with piston speed a constant, fuel-air ratio a constant, and varying values of p_s/p_e.

4. Investigation of the flow coefficients of typical inlet and exhaust ports. This investigation indicated that considerable improvement in output could be obtained by improvement in the design of engine porting.

APPENDIX A

SAMPLING CHECK VALVE

A scaled drawing and exploded-view photograph of the sampling check valve is appended. This valve was developed by applying design changes to the original sampling valve used by the thesis group of Waldron, Keller, Irish, and Karl in their thesis "Measurement of Scavenging Efficiency of the Two-Stroke Engine: A Comparison and Analysis of Methods."

The success of the sampling valve depends on its ability to pass exhaust products from the cylinder, and to exclude scavenging air. This was accomplished by maintaining a pressure behind the check valve (sampling pressure) slightly higher than the scavenging air pressure. The valve was designed with line contact between the seat and check valve. This design made the area of the check valve exposed to blow-down approximately equal to the area exposed to sampling pressure. In this manner multiplying effects of valve area and pressure were eliminated. The check valve was spring loaded to prevent chatter, and to assist the initial building up of back pressure. The check valve was designed so that travel was positive stop-limited to 1/4 the diameter of the seating area. The valve was lapped in to a tight seat before use. The leakage was not important except when it was large enough to materially lower the sampling pressure.

As has been mentioned above, the valve described was the result of changes made as the investigation advanced. The original valve had a water-jacketed tip for cooling. The cooling water lines to the tip increased the overall diameter of the valve tip. The spaces in which the valve was placed were confined, and the tip had to be offset from the center line of the collecting tube. The cooling water lines prevented location of the valve in this position. It was noticed that cutting off the cooling water seemed to have little effect on the valve tip operation during the running of the four-stroke engines. For these reasons, the cooling lines to the valve tip were removed. Cooling of the tip was accomplished by complete water jacketing of the collecting tube leading to the tip.

In addition to the change in cooling, the valve tip had to be redesigned to get sufficient sampling pressure with the reduced blow-down pressure of the compression ignition engines. The entrance hole at the tip of the original valve was very small. This hole was enlarged to give the gases a better chance to strike the valve plunger. The original valve had a ball seat. It was difficult to keep this seat tight because the valve guide had been made loose to prevent sticking at high temperatures. A new valve was manufactured with close guide clearances, (approximately .002" to .003" in diameter) and a 20° angle tip. The seat was made 30°. This arrangement had some guiding value, and at the same time a very narrow line

of contact could be established.

The original valve tip was screwed on the base with 1/8" pipe thread. The position of the tip on the base was critical since it controlled the valve travel. Such an arrangement meant that travel was a function of the degree of tightness of the tip. Further, wear of the threads reduced travel of the valve. This was changed by making the threads 5/16" N.F.T. and using a brass spacer washer to make the joint between tip and base tight. The valve length, the positive stop, and the brass washer were all machined so that the correct travel was given the valve upon assembly. This valve, as redesigned, operated satisfactorily on the F.M. engine. Proper location of the valve with water jacketed collecting tube was impossible on the G.M. 3-71. The valve tip had to be located by trial and error. This required continual bending of the collecting tube. The jacketed collecting tube could not stand this bending. The copper, water jacketed collecting tube was replaced by a 1/4" steel collecting tube with no provision for cooling. The valve functioned properly with no coolant. This valve (see Plate IV) was used on the G.M. 3-71 and M.I.T. ICF. C.F.R. engines.

It was noted upon disassembling this valve, that the spring had been permanently distorted to the point where it was the same length as the positive stop on which it was mounted. Obviously it had no effect on the valve during

operation. From this it may be concluded, that except for the initial build up of pressure and the prevention of chatter initially the spring had no use. It is believed that if the back pressure were built up by compressed air initially, the spring could be eliminated.

An effort was made to develop a steel bodied, external check valve with a steel ball valve. It was reasoned that it might be mounted on a small copper probe external to the manifold and permit better location of the probe in the exhaust blow-down stream.

The ball proved a poor check. An aluminum plunger with a rubber O ring gasket was substituted. This valve built up a good sampling pressure. In actual use on the Fairbanks Morse engine it collected samples which were very highly contaminated by scavenging air. This was probably due to dynamic effects in the probe. The valve was abandoned after its first trial.

APPENDIX B

THE ORSAT APPARATUS

The Orsat apparatus first used was the familiar three-bubbler type for analyzing flue gas. This apparatus was later supplemented by a Hays Gas Analyzer. The principles of operation are the same. The latter is not thought to be as accurate, because the sample analyzed is only 50 cc compared to 100 cc in the bubbler type. However, in actual operation, the two units compared well within the total accuracy limits of the investigation.

It was decided first to rely on commercial chemical reagents for the absorption of the gases. Trouble was discovered immediately with the reagent OXSORBENT, manufactured by the Burrell Corp., Pittsburgh, Pennsylvania.

The procedure was to subject the gas sample to the potassium-hydroxide (KOH) solution (330 gms to 1,000 cc, distilled water). The decrease in volume was then measured as is customary. Further exposure to the OXSORBENT eliminated the O_2 in the gas sample. When the sample was measured for decrease in volume and then rechecked after further exposure to the reagent, it was found that the volume increased rather than decreased. The apparatus was checked for leaks and found to be tight. Further the gas sample was found to be held at the same temperature during the testing. Apparently the OXSORBENT

gave off gases when it was exposed to the exhaust gas even though it did absorb the O_2. Nitrogen was bubbled through the ABSORBENT to drive off any such gases, but the results were the same. A solution of 20 gms pyrogallic acid crystals in 45 cc of water and 90 cc of potassium-hydroxide solution was used for O_2 absorption thereafter with excellent results. COSORBENT was used to extract the CO with success. The only difficulty with this reagent is that it precipitates after use and will plug the holes in the bubbler.

APPENDIX C

AIR FLOW

Since the measurement of air flow by use of an ASME Square Edged Orifice with flange taps used in this thesis is widely known and used in experimental laboratories, it is not felt necessary to discuss the method here.

The M.I.T. reader is referred to a set of notes published by the Sloan Laboratory, M.I.T., entitled - "The Measurement of Air Flow by Means of the ASME Square Edged Orifice with Flange Taps", by William A. Leary. The equations and methods of computation of air flow used in this thesis are discussed in detail on pages 13 through 14 of the above notes.

The information in the above notes is based on material derived from the following publications on fluid meters:

(1) History of Orifice Meters and the Calibration, Construction, and Operation of Orifices for Metering. The American Society of Mechanical Engineers, New York, 1935.

(2) Flow Measurement by Means of Standardized Nozzles and Orifice Plates, ASME Power Test Codes, Part V, Chapter 4. The American Society of Mechanical Engineers, New York, 1940.

(3) Fluid Meters: Their Theory and Application, 4th Edition. The American Society of Mechanical Engineers, New York, 1937.

APPENDIX D

Included in this section are the smooth data sheets, showing results of data and calculations for all points shown on curves of Figures 7 through 11.

DATA: CFR ENGINE, SPARK IGNITION WITH PREMIXING TANK

RPM = 1200
C.R. = 6
BPSA
$P_t = -2"$ Hg
$P_e = +2"$ Hg

DATE	RUN	P_a	ΔP_1	P_1	P_1	Δh	Δh	$\Delta h/P_1$	Y_1	T_1	M	R_e	K/K_m	M_s	w_F	F	% VOL CO_2	% VOL CO	% VOL O_2
		"Hg abs	"Hg	"Hg abs	#/in² abs	"water	#/in²			°F abs	#/sec			#/sec	#/sec				
12-7-49	1	29.67	-1.30	28.07	13.8	7.07	.25	.018	.994	528	.01111	2.66	.997	.01108	.00083	.075	10.8	2.8	.20
12-10-49	1	30.60	-1.30	29.30	14.4	7.3	.26	.018	.994	542	.01148	2.72	.9975	.01145	.000905	.079	11.15	4.5	
	2	30.55	-1.25	"	"	"	"	"	"	545	.01145	"	"	.01141	"	.0791	11.4	4.6	
12-17-49	1	30.6	-1.25	29.35	14.42	7.45	.27	"	"	543	.01160	2.79	.998	.01157	.00072	.0621	12.8	0.8	1.2
	2	30.5	-1.30	29.20	14.38	7.55	"	.019	"	548	"	"	"	.01157	.000673	.0581	13.1		2.3
12-20-49	1	30.2	-1.35	28.85	14.20	7.10	.26	.018	"	546	.01118	2.70	.9975	.01115	.0008	.0718	12.2	1.8	
	2	"	"	"	"	7.15	"	"	"	548	.0112	2.70	"	.01118	.000765	.0685	12.4		0.10
	3	30.15	-1.30	"	"	7.20	"	"	"	548+	.01122	"	"	.01120	.000735	.0656	13.8	.80	.20
	4	30.10	"	28.80	14.16	7.25	"	"	"	549	.01124	"	"	.01123	.000688	.0611	13.9		1.10
	5	30.05	"	28.75	14.15	7.15	"	"	"	550	.01118	"	"	.01115	.000723	.0648	14.0	.30	.40
	6	30.03	-1.27	28.76	14.15	7.10	"	"	"	"	.01112	"	"	.0111	.00075	.0675	13.9	1.1	.10
	7	30.00	-1.30	28.70	14.13	7.35	"	"	"	"	.01131	2.72	"	.01129	.000618	.05475	12.4		3.1
	8	"	"	"	"	7.25	"	"	"	"	.01122	2.70	"	.0112	.00066	.0589	13.0		2.0
	9	"	"	"	"	7.37	"	"	"	"	.01131	2.72	"	.01129	.000585	.0518	12.9		3.1
12-28-49	1	30.08	-1.30	28.78	14.15	7.25	.26	.018	.994	541	.01130	"	"	.01128	.000765	.068	13.4	.3	1.3
	2	"	"	"	"	7.30	"	"	"	544	.01132	"	"	.0113	.00075	.0664	14.1	.8	.3
	3	"	"	"	"	"	"	"	"	"	"	"	"	.0113	.00072	.0637	14.1		.5
	4	"	"	"	"	7.35	"	"	"	545	.01136	2.73	"	.01134	.000701	.0618	14.0		1.0
	5	"	"	"	"	7.45	.27	.019	"	544	.01142	2.75	"	.0114	.00068	.0597	13.7		1.5
	6	"	"	"	"	7.55	"	"	"	"	.01153	2.78	.998	.01151	.000643	.0558	13.0		1.6
	7	30.10	"	28.80	14.16	7.65	"	"	"	"	.01161	2.80	"	.01159	.000605	.0523	12.5		3.3
	8	"	"	"	"	"	"	"	"	"	"	"	"	"	.000615	.0531	12.0		4.0
	9	"	"	"	"	7.25	.26	.018	"	"	.01131	2.72	.9975	.01129	.000772	.0685	12.7	.2	2.3

DATA: MODIFIED CFR ENGINE (COMET HEAD), COMPRESSION IGNITION

RPM = 1400
$P_i = -2" Hg$
$P_e = +2" Hg$

DATE	RUN	P_a	ΔP_1	P_1	T_1	Δh	M_{AS}	FLOW CORRECT.	M_a	ω_F	F	% VOL CO_2	% VOL O_2	BL	FL
		"Hg abs	"Hg	"Hg abs	°F abs	"water	#/sec		#/sec	#/sec $\times 10^{-6}$				"Hg	"Hg
1-4-50	1	29.86	-.05	29.81	538	9.6	.0135	.98	.01323	745	.0563	12.2	3.8		
	2	"	"	"	"	"	"	.98	"	633	.0478	10.8	5.8	10.2	8.3
	3	"	"	"	537	9.85	.0137	"	.01342	565	.0421	9.3	5.7	7.9	8.3
	4	"	"	"	"	9.9	.01372	"	.0134	450	.0336	8.1	9.6	4.6	"
1-31-50	1	30.35	-.05	30.30	533	8.7	.01285	.993	.01278	645	.0505	11.0	5.8	9.1	8.8
	2	"	"	"	"	8.73	.01288	"	.0128	680	.0531	11.2	5.0	9.9	
	3	"	"	"	"	8.70	.01285	"	.01278	705	.0551	12.6	2.5	10.4	
	4	"	"	"	"	"	"	.998	"	737	.0577	12.2	4.0	10.9	8.5
	5	"	"	"	"	9.05	.0131	.993	.01302	400	.0307	6.3	11.9	3.2	
	6	30.35	"	"	"	"	"	"	"	425	.0326	6.8	11.4	3.9	8.7
	7	30.25	-.05	30.20	534	8.85	.01295	.989	.0128	495	.0388	8.4	10.0	4.8	8.8
	8	"	"	"	"	8.90	.0130	"	.01285	512	.0398	8.5	9.0	5.5	
	9	"	"	"	530	9.0	.01308	.993	.013	470	.0362	6.4	10.8	4.2	
	10	"	"	"	"	8.9	.013	"	.01292	540	.04175	8.6	8.9	6.2	
	11	"	"	"	531	8.85	.01295	.992	.01284	575	.0448	9.8	7.4	7.1	
	12	"	"	"	"	"	"	"	"	600	.0467	9.9	6.9	7.9	8.7
2-2-50	1	30.60	-.05	30.55	534	8.95	.01305	.996	.013	425	.0327	6.2	11.8	2.9	8.8
	2	"	"	"	"	8.90	.0130	"	.01295	440	.0341	6.2	11.3	3.6	8.8
	3	"	"	"	"	8.85	.01296	"	.0129	470	.0365	7.3	10.9	4.3	
	4	"	"	"	533	8.80	.01293	"	.01289	493	.0382	8.0	10.0	5.2	
	5	"	"	"	534	8.75	.0129	"	.01286	522	.0406	8.5	9.0	6.2	
	6	"	"	"	"	"	"	"	"	555	.043	8.9	8.4	7.0	8.8
	7	"	"	"	"	9.05	.0131	"	.01304	367	.0281	5.6	13.2	1.8	8.8
	8	"	"	"	531	9.00	.01308	"	.01301	415	.0319	6.6		2.9	

DATA: MODIFIED CFR ENGINE (COMET HEAD), COMPRESSION IGNITION

RPM = 1400
P_i = -2"Hg
P_e = +2"Hg

DATE	RUN	P_a "Hg abs	ΔP_i "Hg	P_i "Hg abs	T_i °F abs	Δh "water	M_{AS} #/sec	FLOW CORRECT.	M_A #/sec	w_F #/sec $\times 10^{-6}$	F	% VOL CO_2	% VOL O_2	BL "Hg	FL "Hg
2-2-50	9	30.60	-.05	30.55	530	9.00	.01308	.996	.01305	380	.0291	5.8			
	10	"		"	534	8.80	.01293	"	.01290	430	.0333	6.6	11.6	3.4	
	11	"		"	"	"	"	"	.01290	450	.0349	7.2	11.0	4.0	
	12	"		"	"	8.85	.01295	"	.01292	475	.0367	7.7	9.8	4.6	8.7
	13	"		"	"	"	"	"	.0129	442	.0342	6.8	11.2	3.6	8.6
	14	30.55	-.05	30.50	"	"	"	.995	.0129	500	.0387	8.0	10.0	5.2	
	15	"		"	"	8.80	.01293	"	.01288	512	.0398	7.8	9.2	5.8	
	16	"		"	"	"	"	"	.01288	528	.041	8.4	8.9	6.2	
	17	"		"	"	"	"	"	"	546	.0424	9.1	8.4	6.8	
	18	"		"	"	8.75	.01288	"	.01280	572	.0447	9.2	8.2	7.6	8.5
	19	"		"	"	8.70	.01285	"	"	590	.0461	9.5	7.5	8.1	8.5
	20	"		"	"	"	"	"	"	610	.0476	10.0	7.0	8.5	
	21	"		"	"	"	"	"	"	647	.0505	10.0	7.3	9.1	
	22	"		"	"	8.65	.01282	"	.01278	665	.0521	11.2	5.3	9.6	
	23	"		"	"	8.60	.01278	"	.01272	705	.0553	11.2	4.6	10.0	
	24	"		"	"	8.55	.01273	"	.0127	744	.0586	12.6	3.7	10.9	8.2
2-5-50	1	30.20	-.05	30.15	534	8.90	.013	.995	.0129	306	.0237	4.7	14.3	0.3	8.5
	2	"		"	"	"	"	.989	.0129	339	.0263	5.1	13.4	1.25	
	3	"		"	"	8.85	.01296	"	.01282	360	.028	5.0	13.5	2.00	
	4	"		"	"	"	"	"	"	389	.0303	5.8	12.6	2.50	
	5	"		"	532	8.90	.0130	.992	.0129	350	.0271	5.2	12.8	1.5	
	6	"		"	532	8.80	.01292	"	.0128	415	.0324			3.0	

DATA: MODIFIED CFR ENGINE (COMET HEAD), COMPRESSION IGNITION

RPM = 1400
$P_t = -2'' Hg$
$P_e = +2'' Hg$

DATE	RUN	P_a	ΔP_i	P_i	T_i	Δh	M_{AS}	FLOW CORRECT.	M_A	W_F	F	% VOL CO_2	% VOL O_2	BL	FL
		"Hg abs	"Hg	"Hg abs	°F abs	"water	#/sec		#/sec	#/sec				"Hg	"Hg
2-5-50	7	30.20	-.05	30.15	532	8.75	.0128	.992	.0127	.00045	.0354	6.0	12.0	3.9	
	8	"	"	"	"	"	"	"	"	.00042	.0332	6.5	12.0	3.3	8.3
	9	"	"	"	"	"	"	"	"	.00047	.037	7.4	10.6	4.1	
	10	"	"	"	"	8.50	.0127	"	.0126	.00081	.0637	13.0	3.1	11.0	8.2
2-8-50	1	29.80	-.10	29.70	532	8.90	.013	.982	.01277	.000495	.0388	7.4	10.1	4.3	9.1
	2	"	-.05	29.75	"	8.85	"	"	"	.000455	.0356	6.1	10.3	3.4	
	3	"	"	"	"	8.90+	.01308	"	.01282	.00039	.0304	6.1	11.9	3.1	
	4	"	"	"	"	9.1	.01317	"	.01292	.00033	.0256	5.2	13.7	1.0	8.9

DATA: FAIRBANKS MORSE OPPOSED PISTON DIESEL ENGINE MODEL # 38E5 ¼

CONTINUED ON NEXT PAGE →

DATE	RUN	P_a "Hg abs.	ΔP_1 "Hg	P_1 "Hg abs	P_1 #/in² abs	Δh " WATER	Δh #/in²	$\Delta h/P_1$	Y_1	T_1 °F abs.	M #/sec	R_e $\times 10^3$	K/K_m	M_s #/sec	w_F #/sec	RPM	VOLTAGE A.C.	AMPS A.C.	t_{i_1} °F	t_{i_2} °F
3-3-50	1	30.15	-0.4	29.75	14.60	26.75	.965	.066	.979	534	1.183	468	.997	1.179	.016	1242	382	129	—	144
	2	"	-13.15	17.00	8.35	12.2	.439	.0526	.981	527	.605	208	1.001	.605	.01078	"	155	67	—	268
	3	"	-0.05	30.10	14.79	7.4	.268	.018	.993	532	.633	218	1.001	.633	.00685	670	250	90	—	104
	4	30.15	-7.8	22.35	11.0	4.65	.1675	.0152	.993	527	.436	150	1.003	.436	.0057	"	200	70	—	164
3-4-50	1	30.45	-0.4	30.05	14.78	26.3	.949	.0642	.98	536	1.176	404	.997	1.174	.016	1242	380	129	—	145
	2	"	-13.2	17.25	8.48	11.9	.429	.0506	.984	529	.606	209	1.001	.606	.0109	"	140	52	298	268
	3	30.45	-8.7	21.75	10.70	18.5	.666	.0623	.98	531	.844	290	.999	.844	.0121	1242	250	90	220	208
3-10-50	1	30.15	-4.15	26.0	12.78	23.5	.856	.067	.978	529	1.040	360	.997	1.037	.01545	1242	358	120	170	155
	2	"	-10.0	20.15	9.90	16.3	.587	.0594	.98	524	.766	264	1.00	.766	.01195	"	235	80	232	199
	3	"	-0.15	30.0	14.75	11.55	.416	.0282	.99	529	.792	272	"	.792	.00838	829	275	95	104	103
	4	"	-2.35	27.80	13.68	10.95	.394	.029	"	531	.741	258	"	.741	"	"	270	92	122	115
	5	"	-4.60	25.55	12.57	9.9	.356	.028	"	527	.678	235	"	.678	"	"	280	95	142	130
	6	"	-7.15	23.00	11.30	8.5	.308	.027	.99	"	.595	204	1.001	.595	.00805	"	270	92	167	148
	7	"	-11.05	19.10	9.38	5.9	.212	.023	.992	530	.451	155	1.004	.453	.00885	"	270	93	224	187
	8	"	-9.05	21.10	10.37	7.15	.257	.025	"	"	.523	180	1.002	.524	.00873	"	255	88	195	170
	9	"	-2.35	27.80	13.70	10.9	.392	.029	.99	532	.737	261	1.00	.737	.00838	"	270	93	124	119
	10	30.15	-2.35	27.80	13.70	11.15	.40	.029	"	"	.747	257	"	.747	"	"	273	93	122	117
3-17-50	1	29.90	-0.35	29.55	14.52	24.40	.882	.061	.98	532	1.128	388	.996	1.123	.01689	1200	405	135	142	133
	2	"	-0.32	29.58	14.53	"	"	"	"	"	"	"	"	"	"	"	400	133	142	134
	3	29.90	"	"	"	"	"	"	"	"	"	"	"	"	"	"	405	138	143	135
3-18-50	1	29.35	-12.50	16.85	8.29	11.2	.403	.049	.983	526	.582	203	1.001	.583	.01205	1200	235	83	289	236
	2	"	"	"	"	11.1	"	"	"	527	.578	"	"	.579	"	"	215	78	290	237
	3	29.35	"	"	"	"	"	"	"	"	"	"	"	"	"	"	220	79	288	236

CONTINUED NEXT PAGE →

DATA: FAIRBANKS MORSE OPPOSED PISTON DIESEL ENGINE MODEL # 38E5¼

CONTINUED NEXT PAGE →

DATE	RUN	P_e	P_2	P_s	t_2	T_{e_1}	T_{e_2}	T_{e_3}	Av. % Vol CO_2	Av. % Vol O_2	F	M_{RET}	P_e	T_{e_2}	ρ_s	e_s	R_s	CYLINDER SAMPLED
		"Hg	"Hg	"Hg	°F	°F abs	°F abs.	°F abs.				#/sec	#/in² abs.	°F abs.	#/ft³			Nº
3-3-50	1	+.70							5.2	13.2	.0274	.584	15.17	604	.0678	.719	1.45	3
	2	+.20							4.8	13.8	.0254	.424	14.90	728	.0553	.642	.915	3
	3	+.20							5.2	13.5	.0268	.256	14.92	564	.0715	.554	1.372	3
	4	+.10							4.2	14.7	.0228	.250	14.85	620	.0647	.598	1.043	3
3-4-50	1	+.70	-2.1	+6.3	72				5.0	13.8	.026	.615	15.30	605	.0683	.751	1.434	3
	2	+.20		+2.4	"				4.5	14.1	.0244	.446	15.05	728	.05575	.667	.906	3
	3	+.40	-9.9	+3.0	70				4.3	14.6	.023	.525	15.15	668	.0613	.715	1.15	3
3-10-50	1	+.50	-5.7	+5.2	66				5.0	14.0	.0257	.601	15.10	615	.0663	.756	1.305	3
	2	+.30	-11.0	+3.6	65				4.0	15.2	.0215	.555	14.97	659	.0613	.756	1.045	3
	3	+.30	-.95	+2.7	64				3.8	15.5	.0206	.407	14.97	563	.0717	.711	1.383	3
	4	+.28	-3.1	+2.7	67				4.0	15.1	.0218	.384	14.96	575	.0702	.685	1.322	3
	5	+.25	-5.25	+2.65	68				3.8	15.4	.0208	.403	14.95	590	.0684	.739	1.242	3
	6	+.18	-7.75	+1.7	66				3.5	15.7	.0199	.404	14.90	608	.0662	.765	1.126	3
	7	+.10	-11.4	+0.5	67				4.7	13.8	.0252	.351	14.89	647	.0621	.708	.914	3
	8	+.12	-10.6	+0.6	"				4.2	14.9	.0224	.39	14.90	630	.0639	.765	1.028	3
	9	+.25	-3.1	+2.2	65				5.1	13.3	.027	.31	14.96	579	.0697	.557	1.328	2
	10	+.25	-3.1	+2.2	63				4.6	14.0	.0248	.338	14.96	577	.070	.604	1.336	1
3-17-50	1	+.62	-2.0	+5.8	71				5.7	12.7	.0291	.580	15.00	594	.0682	.735	1.423	3
	2	"	"	"	"	460		480	5.8	12.9	.029	"	"	"	"	"	"	2
	3	"	"	+5.85	"		490	485	5.8	12.7	.0293	"	"	"	"	"	"	1
3-18-50	1	+.20	-13.2	+2.15			660	640	5.8	12.4	.0298	.404	14.53	.696	.0563	.62	.895	1
	2	"	"	+2.10		615	675		5.6	12.9	.0284	.424	14.52	697	.0562	.651	.890	3
	3	"	"	"		610		630	6.1	12.3	.0306	.394	14.53	696	.0563	.604	.888	2

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DATA: FAIRBANKS MORSE OPPOSED PISTON DIESEL ENGINE MODEL # 38E5¼

DATE & RUN	ORSAT #1 CO_2	ORSAT #1 O_2	ORSAT #2 CO_2	ORSAT #2 O_2
3-3-50 - 1	5.1	13.7	5.2	13.2
	5.0	13.6	5.2	13.3
	4.9	12.4	5.4	13.2
			5.3	13.3
	av. CO_2 = 5.2			
	O_2 = 13.2			
2	4.6	13.9	5.0	13.6
	5.0	14.0	4.8	13.8
			4.8	13.8
	av. CO_2 = 4.8		4.8	13.8
	O_2 = 13.8			
3	5.3	13.5	5.0	13.4
	5.3	13.6	5.6	13.4
			5.0	13.6
	av. CO_2 = 5.2		5.1	13.5
	O_2 = 13.5			
4	4.2	15.0	4.5	14.0
	4.0	14.8	4.2	14.8
			4.0	14.9
			4.0	14.8
3-4-50 - 1			5.0	13.6
	av CO_2 = 5.0		5.0	13.8
	O_2 = 13.8		4.8	13.8
2	av CO_2 = 4.5		4.6	14.0
	O_2 = 14.1		4.4	14.2
3			4.3	14.5
	av CO_2 = 4.3		4.2	14.7
	O_2 = 14.6		4.3	14.5
			4.4	14.6
3-10-50 - 1	5.0	14.0	5.0	13.7
	4.8	14.0	4.9	13.9
			5.0	14.1
	av CO_2 = 5.0			
	O_2 = 14.0			
2	4.2	14.8	4.4	14.5
	3.8	15.2	4.0	15.4
	4.0	15.5	3.8	15.4
			3.8	15.4
	av CO_2 = 4.0			
	O_2 = 15.2			

DATE & RUN	ORSAT #1 CO_2	ORSAT #1 O_2	ORSAT #2 CO_2	ORSAT #2 O_2
3-10-50-3	3.8	15.6	3.6	15.3
	3.9	15.5	3.7	15.9
	av. CO_2 = 3.8		4.0	14.9
	O_2 = 15.5		3.8	15.6
4	3.8	15.2	4.2	14.8
	4.0	15.1	4.1	15.2
			4.0	15.1
	av. CO_2 = 4.0			
	O_2 = 15.1			
5	3.7	15.7	3.8	15.4
	3.8	15.4	3.8	15.4
			4.0	15.4
	av. CO_2 = 3.8			
	O_2 = 15.4			
6	3.6	15.8	3.2	15.8
	3.7	15.5	3.7	15.7
			3.4	15.8
	av. CO_2 = 3.5			
	O_2 = 15.7			
7	4.6	13.9	4.8	13.6
			4.7	13.7
	av. CO_2 = 4.7		4.6	13.8
	O_2 = 13.8			
8	4.3	14.9	4.1	14.9
	av. CO_2 = 4.2		4.1	14.8
	O_2 = 14.9			
9	av CO_2 = 5.1		5.1	13.5
	O_2 = 13.3		5.1	13.1
10	4.7	14.5	4.5	13.9
			4.6	13.6
	av. CO_2 = 4.6			
	O_2 = 14.0			

DATE & RUN	ORSAT #1 CO_2	ORSAT #1 O_2	ORSAT #2 CO_2	ORSAT #2 O_2
3-17-50-1	5.3	13.3	5.6	11.6
	5.7	12.9	5.6	12.8
	5.7	12.9	6.0	12.5
			5.9	12.7
	av. CO_2 = 5.7		5.8	12.7
	O_2 = 12.7			
2	5.7	12.9	5.6	13.0
	6.0	12.8	5.9	12.7
	av. CO_2 = 5.8		5.6	12.9
	O_2 = 12.9		5.8	12.8
			5.8	12.9
3	5.6	12.8	5.8	12.8
	5.8	12.6	5.8	12.8
	av. CO_2 = 5.8		5.8	12.7
	O_2 = 12.7		6.0	12.6
3-18-50-1			5.8	12.0
	av. CO_2 = 5.8		5.8	12.6
	O_2 = 12.4		5.7	12.5
			5.8	12.4
2			6.0	12.6
	av CO_2 = 5.6		5.2	13.0
	O_2 = 12.9		5.6	13.0
			5.5	13.1
3			6.2	12.1
	av. CO_2 = 6.1		6.2	12.2
	O_2 = 12.3		6.2	12.2
			6.0	12.8

DATA: GENERAL MOTORS SERIES "71" DIESEL ENGINE MODEL # 3-71

DATE	RUN	P_a	ΔP_1	P_1	P_1	Δh	Δh	$\Delta h/P_1$	Y_1	T_1	M	R_e	K/K_m	M_s	w_f	RPM	VOLTAGE	AMPS.	BL	t_i
		"Hg abs.	"Hg	"Hg abs	#/in^2 abs	"WATER	#/in^2			°F abs	#/sec	$\times 10^3$		#/sec	#/sec		D.C.	D.C.	"Hg	°F
4-7-50	1	30.01	-0.10	29.91	14.71	19.3	.695	.047	.985	544	.2955	183	.998	.295	.00982	1740	132	350	28.0	203
4-8-50	1	29.86	-0.10	29.76	14.64	19.2	.692	.047	.985	541	.2945	183	.998	.294	.00985	1740	132	350	28.2	203
	2	"	"	"	"	"	"	"	"	542	"	"	"	"	"	"	"	"	28.0	"
	3	"	"	"	"	"	"	"	"	"	"	"	"	"	.01085	"	"	"	28.2	202
	4	29.86	-0.10	29.76	14.64	19.15	.690	.047	.985	543	.294	183	.998	.2935	.01075	1740	132	350	28.2	204
4-11-50	1	29.85	-0.10	29.75	14.64	19.05	.685	.047	.985	544	.293+	183	.998	.293	.01035	1740	133	350	28.2	204
4-14-50	1	29.68	-.05	29.63	14.60	6.2	.223	.015	.994	542	.1686	106	1.001	.1688	.00675	1740	87	250	6.8	+1.8

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DATA: GENERAL MOTORS SERIES "71" DIESEL ENGINE MODEL # 3-71

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DATE	RUN	P_e	P_2	P_3	t_2	MINIMUM SAMPLING PRESSURE	AV. % VOL CO_2	AV. % VOL O_2	F	M_{EET}	P_e	T_e	ρ_s	e_s	R_s	CYLINDER SAMPLED	
		"Hg	"Hg	"Hg	°F	"Hg				#/sec	#/in² abs	°F abs	#/ft³			No	
4-7-50	1	+4.70	-3.85	+11.0	82		10.5	6.0	.0503	.195	17.07	663	.0695	.744	1.125	1	
4-8-50	1	+4.70	-3.80	+10.8	79		9.2	7.9	.0443	.222	17.0	663	.0692	.842	1.115	2	
	2	+4.80	-3.85	+10.8	80		10.7	5.6	.0514	.1915	17.07	"	.0695	.724	1.110	3	
	3	+4.80	-3.85	+10.9	81		11.0	5.6	.052	.2087	"	662	"	.788	1.110	1	
							9.3	8.2	.044	.2465	"	"	"	.932	1.110	2	TAKEN UNDER CONDITIONS OF RUN #4
	4	+4.80	-3.85	+10.9	82		10.6	5.9	.0506	.212	"	664	.0694	.812	1.12	3	
4-11-50	1	+4.60	-3.95	+10.5	82	11.0	10.7	5.5	.0518	.20	16.98	664	.0690	.769	1.128	1	
						"	11.4	4.7	.0542	.191	"	"	"	.735	1.128	2	} TAKEN UNDER CONDITIONS OF RUN #1
						"	11.0	5.1	.0528	.196	"	"	"	.745	1.127	3	
4-14-50	1	+1.8	-12.9	+3.1	76	3.5	10.3	5.9	.0502	.1343	15.50	692	.0605	.586	.736	1	
						"	10.1	6.4	.488	.1383	"	"	"	.603	.736	2	} TAKEN UNDER CONDITIONS OF RUN #1
						"	"	6.0	.496	.136	"	"	"	.593	.735	3	

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D-10

DATA: GENERAL MOTORS SERIES "71" DIESEL ENGINE MODEL # 3-71

DATE	RUN & CYL.	ORSAT #1		ORSAT #2	
		CO_2	O_2	CO_2	O_2
4-7-50	1-1			10.6	5.8
		av. CO_2 = 10.5	O_2 = 6.0	10.5	5.9
				10.4	6.0
4-8-50	1-2	9.5	7.6	9.3	7.7
		9.2	8.0	9.1	8.0
				8.9	8.3
		av CO_2 = 9.2	O_2 = 7.9	9.2	8.0
	2-3	10.3	5.2	11.0	5.1
		11.0	5.4	10.7	6.5
				10.8	5.8
		av CO_2 = 10.7	O_2 = 5.6		
	3-1	11.0	5.6	11.0	5.5
		11.2		10.9	5.6
		av CO_2 = 11.0	O_2 = 5.6		
	3-2	8.2	9.1	9.6	7.8
		9.8	7.2	8.4	9.2
		9.7	7.2	8.6	8.8
				9.1	8.5
		av. CO_2 = 9.3	O_2 = 8.2	10.3	6.7
				10.2	6.9
	4-3			10.6	5.8
		av CO_2 = 10.6	O_2 = 5.9	10.4	6.1
				10.9	5.7

DATE	RUN & CYL.	ORSAT #1		ORSAT #2	
		CO_2	O_2	CO_2	O_2
4-11-50	1-1	10.5	—	11.0	5.3
		10.4	—	10.6	5.7
		av CO_2 = 10.7	O_2 = 5.5	10.8	5.5
				10.6	5.6
	1-2			11.6	4.7
		av CO_2 = 11.4	O_2 = 4.7	11.2	4.8
				11.4	4.6
				11.5	4.5
	1-3			10.9	5.2
		av CO_2 = 11.0	O_2 = 5.1	11.1	5.1
				11.1	5.1
4-14-50	1-1	10.0	6.2	10.4	6.1
		10.0	5.7	10.4	5.9
		av CO_2 = 10.3	O_2 = 5.9	10.4	6.0
				10.4	5.6
	1-2	9.7	6.9	10.3	6.6
		10.0	5.9	10.0	6.4
		av CO_2 = 10.1	O_2 = 6.4	10.4	6.2
				10.2	6.3
	1-3	9.6	6.6	10.0	6.4
		10.0	5.8	10.6	5.6
		av CO_2 = 10.1	O_2 = 6.0	10.2	5.9
				10.4	5.5

DATA: MIT (CFR) 2-STROKE, LOOP SCAVENGING, GASOLINE ENGINE

DATE	RUN	P_a	ΔP_1	P_1	P_1	Δh	Δh	$\Delta h/P_1$	Y_1	T_1	M	R_e	K/K_m	M_s	w_F	RPM	VOLTAGE	AMPS	t_i	P_e
		"Hg abs	"Hg	"Hg abs	#/in² abs	"water	#/in²			°F abs	#/sec	X10³		#/sec	#/sec		D.C.	D.C.	°F	"Hg
4-19-50	1	30.01	+9.05	39.06	19.2	12.9	.465	.024	.992	541	.0735	87	.999	.0734	.00272	1930	165	54	83	+0.35
4-20-50	1	29.70	+4.55	33.25	15.86	6.2	.223	.014	.995	538	.0474	56	1.001	.0474	.00206	1930	126	43	81	+0.20
	2	"	+5.4	35.10	17.30	7.65	.275	.016	"	537	.0542	64	1.00	.0542	.0023	"	144	49	80	+0.20
	3	"	+9.05	38.75	18.56	13.1	.471	.027	.992	"	.0742	87.6	.999	.0741	.00271	"	165	55	81	+0.25

DATE	RUN	P_s	MINIMUM SAMPLING PRESSURE	Av. % Vol CO_2	Av. % Vol O_2	Av. % Vol CO	F	M_{RET}	P_e	T_ℓ	ρ_s	e_s	R_s
		"Hg	"Hg					#/sec	#/in² abs	°F abs	#/ft³		
4-19-50	1	7.6	9.0	12.8	0.2	1.9	.0716	.038	14.93	543	.0742	.645	1.243
4-20-50	1	4.0	5.0	12.8	0.4	1.2	.0704	.0292+	14.72	541	.0734	.488	0.79
	2	4.7	6.0	12.4	0.2	1.9	.0723	.0318+	"	540	.0736	.528	0.90
	3	7.85	9.0	12.8	0.2	1.5	.0708	.0383	14.75	541	.0735	.638	1.236

[illegible]

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
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DATA: MIT (CFR) 2 STROKE, LOOP SCAVENGING, GAS ENGINE

DATE	RUN	ORSAT #1			ORSAT #2		
		CO_2	O_2	CO	CO_2	O_2	CO
4-19-50	1	12.8	0	—	12.7	0.5	2.0
		12.7	0	—	13.2	0.1	1.7
		av	CO_2 = 12.8		13.0	0	1.8
			O_2 = 0.2		12.6	0.2	2.1
			CO = 1.9		12.7	0	2.0
4-20-50	1	12.9	0.1	1.2	13.0	0.2	0.9
		12.5	0.2	0.9	12.7	0.8	0.8
		12.6	0.2	2.4	12.9	0.4	1.1
			CO_2 = 12.8		12.6	0.8	1.3
		av.	O_2 = 0.4		13.0	0.7	0.7
			CO = 1.2				
	2	12.4	0	1.8	12.2	0.1	2.1
		12.4	0	1.8	12.5	0.3	2.0
					12.4	0.2	2.1
			CO_2 = 12.4		12.4	0.2	1.6
		av	O_2 = 0.2				
			CO = 1.9				
	3	12.8	0.1	1.6	12.8	0	2.0
		12.7	0.1	1.6	13.0	0.2	1.3
					13.0	0.2	1.4
					12.8	0.3	1.3
			av CO_2 = 12.8				
			O_2 = 0.2				
			CO = 1.5				

APPENDIX E

SAMPLE CALCULATIONS

The sample calculations break down into five separate sections:

A. Calculation of fuel-air ratio of the C.F.R., Spark-Ignition, Engine.

B. Calculation of fuel-air ratio of the modified C.F.R. Engine with Comet Head.

C. Calculation of air supplied, air retained, scavenging efficiency and scavenging ratio of the Fairbanks Morse, Opposed Piston, Engine, Model #38E5¼.

D. Calculation of air supplied, air retained, scavenging efficiency and scavenging ratio of the General Motors, Series "71", Engine, Model #3-71.

E. Calculation of air supplied, air retained, scavenging efficiency, and scavenging ratio of the M.I.T. (C.F.R.), Loop Scavenged Engine.

Sections A and B involve the measurement of fuel rate and the calculation of air supplied to the engine. Flow rate of air was measured by means of an ASME square-edged orifice with flange taps. Fuel rate was measured by use of a rotameter.

Sections C through E involve the calculation of scavenging efficiency and scavenging ratio, where:

$$E_s = \frac{M_{ret}}{NV_c \rho_s}$$

$$R_s = \frac{M_s}{NV_c \rho_s}$$

$$\rho_s = \frac{mp_e}{RT_i}$$

As defined in -

The Internal Combustion Engine

by C. F. Taylor and E. S. Taylor.

where:

M_{ret} = air retained, $^{\#}/_{min.}$

M_s = air supplied, $^{\#}/_{min.}$

N = revol. per minute

V_c = cylinder volume, ft.^{3}

ρ_s = scavenging density, $^{\#}/_{ft.^3}$

m = molecular weight of air, 28.97 $^{\#}/_{\#\ mol.}$

R = 1,544 $\frac{ft.\#}{\#\ mol.\ ^{\circ}R}$

T_i = inlet temp. to cylinder, °R

p_e = exhaust pressure, $^{\#}/_{ft.^2}$

The amount of air retained in the cylinder was established by an analysis of the exhaust gases from which fuel-air (F) could be found. (See Figure 7.)

Since fuel consumption (w_F) was measured by a rotameter, it was then possible to calculate the amount of air retained, M_{ret}

$$\left(\frac{\#}{sec}\right) = \frac{w_F\left(\frac{\#}{sec}\right)}{F}.$$

Air supplied was measured by the same method used in Sections A and B.

All manometer heights, temperatures, etc., necessary for the following calculations are recorded and can be found in the enclosed data sheets. (See Appendix F.)

SECTION A (C.F.R. Engine)

$$M = 0.1145 \times (D_2)^2 \times K_m Y_1 \sqrt{\frac{p_1}{T_1} \Delta h}$$

$$K_m = 0.6043 \text{ where } \beta = \frac{D_2}{D_1} = \frac{.515}{2.067}$$

$$\therefore M = .01837\, Y_1 \sqrt{\frac{p_1}{T_1} \Delta h}$$

Run #1 of 12-7-49

Data: $p_a = 29.07"$ Hg. abs. $\qquad w_F = .0083\ \#/\text{sec.}$

$p_1 = -1.30"$ Hg.

$\Delta h = 7.07"$ water

$T_1 = 528°R$

$p_1 = p_a + \Delta p_1 = 29.07 - 1.00 = 28.07"$ Hg. abs. $= 13.8\#/\text{in}^2$ abs.

$$\Delta h = \frac{7.07" \text{ water} \times 5.188 \dfrac{\#/\text{ft.}^2}{"\text{water}}}{144 \dfrac{\text{in.}^2}{\text{ft.}^2}} = .25\ \#/\text{in.}^2$$

$$\frac{\Delta h}{p_1} = \frac{.25}{13.8} = .018$$

$\therefore Y_1 = .994$ (See Figure 6.)

$$M = .01837 \times .994 \sqrt{\frac{28.07}{528} \times 7.07} - .01111\ \#/\text{sec.}$$

$$R_e = \frac{15.28\, M}{D_2\, \mu} = \frac{15.28 \times .01111}{.515 \times 12.2 \times 10^{-6}} = 2.66 \times 10^4$$

$\therefore K/K_m = .997$ (See Figure 1.).

$$M_s = K/K_m \times M = .997 \times .01111 = .01108\ \#/\text{sec.}$$

$$F - \frac{w_F}{M_s} = \frac{.00083}{.01108} = .075$$

SECTION B C.F.R. Engine with Comet Head

Since it was thought that the "Comet Head" engine would be run over wide ranges of air flow, Figure 2 was set up to facilitate the measurement of air flow.

Run #5 of 1-31-50

Data: $p_a = 30.35"$ Hg. abs. $w_F = .0004$ #/sec.

$\Delta p_1 = -.05"$ Hg.

$\Delta h = 9.05"$ water

$T_1 = 533^{\circ}R$

For $\Delta h = 9.05$, from Figure 2, $M_{AS} = .0131$ #/sec.

$$p_1 = p_a + \Delta p_1 = 30.35 - .05 = 30.30" \text{ Hg.}$$

$$\text{Flow corr.} = \sqrt{\frac{p_1}{29.92} \times \frac{518.4}{T_1}} = \sqrt{\frac{30.30}{29.92} \times \frac{518.4}{533}} = .993$$

$$M_A = M_{AS} \times \text{Flow corr.} = .0131 \times .993 = .01302 \text{ \#/sec.}$$

$$F = \frac{w_F}{M_A} = \frac{.0004}{.01302} = .0307$$

SECTION C Fairbanks Morse Opposed Piston Engine

$$M = 0.1145 \times (D_2)^2 \times K_m \, Y_1 \sqrt{\frac{p_1}{T_1} \Delta h}$$

$$K_m = 0.6519 \text{ where } \beta = \frac{D_2}{D_1} = \frac{3.639}{6.065} = .6$$

$$\therefore \quad M = 0.987 \, Y_1 \sqrt{\frac{p_1}{T_1} \Delta h}$$

Run #1 of 3-17-50, Cylinder #1

Data: p_a = 29.90" Hg. abs.	w_F = .0189 #/sec.
Δp_1 = −0.35" Hg.	RPM = 1200
Δh = 24.40" water	t_{i_2} = 133° F
T_1 = 532° R	p_e = + 0.62" Hg.
C.R. = 17.4	p_s = + 5.8" Hg.

$$p_1 = p_a + \Delta p_1 = 29.90 - .35 = 29.55\text{" Hg. abs.}$$

$$= 14.52 \#/_{in^2}$$

$$\Delta h = \frac{24.40\text{" water} \times 5.88 \#/\text{ft}^2/\text{"water}}{144} = 0.882 \#/\text{in}^2$$

$$\Delta h/p_1 = \frac{.882}{14.52} = .061, \text{ from Figure 6 } Y_1 = .98$$

$$M = 0.987 \times .98 \sqrt{\frac{29.55}{532} \times 24.4} = 1.128 \#/_{sec.}$$

$$R_e = \frac{15.28M}{D_2 \; \mu} = \frac{15.28 \times 1.128}{3.639 \times 12.2} \times 10^{-6} = 388 \times 10^3$$

$$\therefore K/K_m = .996 \text{ (See Figure 3.)}$$

$$M_s = M \times K/K_m = 1.128 \times .996 = 1.123 \#/_{sec.}$$

Analyses of Exhaust Gases gave $CO_2 = 5.7\%$ volume

$O_2 = 12.7\%$ volume

from Figure 7, $F = .0291$

$$M_{ret} = \frac{w_F}{F} = \frac{.01689}{.0291} = 0.58 \ \#/\text{sec.}$$

$$p_e = p_a + p_e = 29.90 + .62 = 30.52'' \text{ Hg.} = 15.00 \ \#/\text{in.}^2$$

$T_{i_2} = 594$ Note: Of the inlet temperatures measured, T_{i_2} was the closest to the inlet ports and it was felt it more closely represented the inlet temperatures.

$$\rho_s = \frac{m \, p_e}{R \, T_i} = \frac{28.97 \frac{\#}{\#\text{mol.}} \times 15.00 \ \#/\text{in.}^2 \times 144 \frac{\text{in.}^2}{\text{ft.}^2}}{1{,}544 \frac{\text{ft.}\#}{\#\text{ mol.}^\circ R} \quad 594^\circ R} = .0682 \frac{\#}{\text{ft}^3}$$

$$V_c = V_d \frac{r}{r-1} = 314 \text{ in.}^3 \times \frac{17.4}{17.4-1} = 999 \text{ in.}^3 = .578 \text{ ft.}^3$$

$$e_s = \frac{M_{ret}}{N V_c \rho_s} = \frac{.58 \ \#/\text{sec.} \times 60 \frac{\text{sec.}}{\text{min.}}}{1200 \frac{\text{revol.}}{\text{min.}} \times .578 \text{ ft.}^3 \times .0682 \ \#/\text{ft.}^3} = .735$$

$$R_s = \frac{M_s}{N V_c \rho_s} = e_s \times \frac{M_s}{M_{ret}} = .735 \times \frac{1.123 \ \#/\text{sec.}}{.58 \ \#/\text{sec}} = 1.423$$

SECTION D General Motors 3-71 Series

$$M = 0.1145 \times (D_2)^2 \times K_m \quad Y_1 \sqrt{\frac{p_1}{T_1} \Delta h}$$

$$K_m = .6266 \text{ where } \beta = \frac{D_2}{D_1} = \frac{2.013}{4.026} = .5$$

$$\therefore M = 0.291 \quad Y_1 \sqrt{\frac{p_1}{T_1} \Delta h}$$

Run #1 of 4-11-50, Cylinder #1

Data: $p_a = 29.85$" Hg. abs.

$\Delta p_1 = -0.10$" Hg.

$\Delta h = 19.05$" water

$T_1 = 544^\circ R$

$w_F = .01035\ \#/\text{sec.}$

$CR = 16$

$RPM = 1740$

$T_i = 664^\circ R$

$P_e = +4.60$" Hg.

$P_s = 10.5$" Hg.

$$p_1 = p_a + \Delta p_1 = 29.85 + -0.10 = 29.75\text{" Hg. abs.} = 14.64\ \#/\text{in.}^2$$

$$\Delta h = \frac{19.05\text{" water} \times 5.88\ \#/\text{ft}^2/\text{" water}}{144\ \text{in}^2/\text{ft.}^2} = .685\ \#/\text{in}^2$$

$$\frac{\Delta h}{p_1} = .047, \text{ from Figure 6, } Y_1 = .985$$

$$M = .291 \times .985 \sqrt{\frac{29.75}{544} \times 19.05} = .293(+)\ \#/\text{sec.}$$

$$R_e = \frac{15.28M}{D_2\ \mu} = \frac{15.28 \times .293}{2.013 \times 12.2 \times 10^{-6}} = 183 \times 10^3$$

from Figure 4 $K/K_m = .998$

$$\therefore M_s = M \times K/K_m = .293\ \#/\text{sec.}$$

Analyses of Exhaust Gases gave $CO_2 = 10.7\%$ volume

$O_2 = 5.5\%$ volume

from Figure 7, $F = .0518$

$$M_{ret} = \frac{\dot{w}_F}{F} = \frac{.01055\ \#/\text{sec.}}{.0518} = 0.20\ \#/\text{sec.}$$

$$P_e = P_a + P_g = 29.85 + 4.60 = 34.45''\ \text{Hg. abs.} = 16.98\ \#/\text{in.}^2$$

$$\rho_s = \frac{m\ P_e}{R\ T_i} = \frac{28.97\ \frac{\#}{\#\text{mol.}} \times 16.98\ \#/\text{in.}^2 \times 144\frac{\text{in.}^2}{\text{ft.}^2}}{1544\ \frac{\text{ft}\ \#}{\#\ \text{mol.}\ ^\circ R}\quad 644\ ^\circ R} = .069\ \#/\text{ft}^3$$

$$V_c = V_d\ \frac{r}{r-1} = 212.7 \times \frac{16}{15} = 226.7\ \text{in.}^3 = .1313\ \text{ft.}^3$$

$$e_s = \frac{M_{ret}}{NV_c\,\rho_s} = \frac{.20\ \#/\text{sec.} \times 60\ \frac{\text{sec.}}{\text{min.}}}{1740\ \text{RPM} \times .1313\ \text{ft.}^3 \times .069\ \#/\text{ft.}^3} = .759$$

$$R_s = \frac{M_s}{NV_c\,\rho_s} = e_s \times \frac{M_s}{M_{ret}} = .759 \times \frac{.293}{.20} = 1.128$$

SECTION E M.I.T. (C.F.R.) Loop Scavenged, Two-Stroke

$$M = 0.1145 \text{ x } (D_2)^2 \text{ x } K_m \ Y_1 \sqrt{\frac{p_1}{T_1} \Delta h}$$

$$K_m = .628 \text{ where } \beta = \frac{D_2}{D_1} = \frac{1.033}{2.067} = .50$$

$$M = .0769 \ Y_1 \sqrt{\frac{p_1}{T_1} \Delta h}$$

Run #3 of 4-20-50

Data: p_a = 29.7" Hg. abs.

Δp_1 = + 9.05" Hg.

Δh = 13.1" water

T_1 = 537° R

w_F = .00271

CR = 6.72

RPM = 1930

T_i = 541

p_e = + 0.35" Hg.

p_s = + 7.85" Hg.

$$p_1 = p_a + \Delta p_1 = 29.7 + 9.05 = 18.56 \ \#/\text{in.}^2$$

$$\Delta h = \frac{13.1\text{" water x } 5.188 \ \#/\text{ft.}^2/\text{" water}}{144 \text{ in.}^2/\text{ft.}^2} = .471 \ \#/\text{in.}^2$$

$$\frac{\Delta h}{p_1} = \frac{.471}{18.56} = .027, \text{ from Figure 6, } Y_1 = .992$$

$$M = .0769 \text{ x } .992 \sqrt{\frac{38.75}{537} \text{ x } 13.1} = .0742 \ \#/\text{sec.}$$

$$R_e = \frac{15.28M}{D_2 \ \mu} = \frac{15.28 \text{ x } .0742}{1.033 \text{ x } 12.2 \text{ x } 10^{-6}} = 87.6 \text{ x } 10^3$$

from Figure 5, $K/K_m = .999$

$$M_s = M \text{x}^{K/K_m} = .0742 \text{ x } .999 = .0741 \ \#/\text{sec.}$$

Analyses of Exhaust Gases $CO_2 = 12.8\%$ volume

$O_2 = 0.2\%$ volume

$CO = 1.5\%$ volume

from Figure 7, $F = .0708$

$$M_{ret} = \frac{w_F}{F} = \frac{.00271}{.0708} = .0383$$

$$p_e = p_a + p_e = 29.71 + .35 = 30.05'' \text{ Hg. abs.} = 14.75\ \#/\text{in.}^2$$

$$\rho_s = \frac{m\,p_e}{R\,T_i} = \frac{28.97 \frac{\#}{\#\text{mol.}} \times 14.75\ \#/\text{in.}^2 \times 144 \frac{\text{in.}^2}{\text{ft.}^2}}{1544 \frac{\text{ft.}\#}{\#\ \text{mol.}^\circ R} \times 541^\circ R} = .0735\ \#/\text{ft.}^3$$

$$V_c = v_c \frac{r}{r-1} = \frac{3.14 \times \overline{3.25}^2 \times 4\frac{1}{2}}{4 \times 1728} \times \frac{6.72}{5.72} = .0254 \text{ ft.}^3$$

$$e_s = \frac{M_{ret}}{N V_c \rho_s} = \frac{.0383 \frac{\#}{\text{sec.}} \times 60 \frac{\text{sec.}}{\text{min.}}}{1930 \text{ RPM} \times .0254 \text{ ft.}^3 \times .0735\ \#/\text{ft.}^3} = .638$$

$$\lambda_s = \frac{M_s}{N V_c \rho_s} = e_s \frac{M_s}{M_{ret}} = .638 \times \frac{.0741}{.0383} = 1.236$$

APPENDIX F

SYMBOLS

(in alphabetical order)

BL - Brake load

CO - Carbon monoxide

CO_2 - Carbon dioxide

CR - Compression ratio

D_1 - Inside pipe diameter

D_2 - Orifice diameter

e_s - Scavenging efficiency $= \frac{M_{ret}}{\rho_s N V_c}$

F - Fuel-air ratio

FL - Friction load

h - Pressure drop across the orifice

Hg - Mercury

H_2O - Water

K_m - Mean flow coefficient

K/K_m - Ratio of flow coefficients

m - Molecular weight, #/#mol.

M - Air rate supplied, uncorrected

M_s - Air rate supplied, corrected

M_A - Air rate supplied, uncorrected, C.F.R. Comet Head Engine.

M_{AS} - Air rate supplied, corrected, C.F.R. Comet Head Engine

M_{ret} - Air retained

N - Engine revolutions per minute

O_2 - Oxygen

p_a - Atmospheric pressure

p_e - Exhaust pressure

p_i - Inlet pressure before the cylinder inlet ports

p_s - Scavenging pressure (pressure after the scavenging blower and before the cylinder inlet ports)

p_1 - Pressure before the orifice

p_2 - Pressure before the scavenging blower

Δp_1 - Pressure drop before the orifice

R - Universal gas constant $= 1544 \frac{ft.\#}{\# \, mol. \, ^{o}R}$

RPM - Engine revolutions per minute

R_s - Scavenging ratio $= \frac{M_s}{\rho_s N V_c}$

R_e - Reynolds number $= \frac{\rho u D_2}{12 \mu}$

$$\cong \frac{15.28 \, w_c}{D_2 \mu} \cong 1.24 \times 10^6 \frac{w_c}{D_2}$$

r - Compression ratio

s - Piston speed $= 2SN$

S - Stroke

T_1 - Temperature before the orifice

T_2 - Temperature before the scavenging blower

T_i - Inlet temperature

T_{i_1} - Temperature in the inlet manifold immediately following the Fairbanks Morse scavenging blower

T_{i_2} - Temperature in the inlet manifold before the inlet ports of the Fairbanks Morse Engine

T_e - Exhaust gas temperature - numeral subscripts refer to each cylinder of Fairbanks Morse Engine

u - Fluid velocity through orifice, ft./sec.

V_d - Cylinder displacement

V_c - Cylinder volume

w_F - Fuel rate

Y_1 - Expansion factor for air (Figure 6)

ρ - Air density before orifice, $\#/\text{ft.}^3$

ρ_s - Scavenging density $= \dfrac{m\ p_e}{R\ T_i}$

μ - Fluid viscosity before the orifice $\dfrac{\#\ \text{mass}}{\text{ft. sec.}}$ = const. at 12.3 x 10^{-6} $\#/$ft. sec. (Air)

" - inches

APPENDIX G

BIBLIOGRAPHY

TEXTS OR PAMPHLETS

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B. Report on Two-Stroke Engines - C. F. Taylor

C. The Compostion of Diesel Exhaust Gas - Elliot and Davis - Report published for presentation at the SAE National Diesel Engine Meeting, November 1 - 2, 1949

D. The Measurement of Air Flow by Means of the ASME Square-Edged Orifice with Flange Taps - William A. Leary - Notes published for Sloan Laboratory, Massachusetts Institute of Technology, Cambridge, Massachusetts

E. Measurement of Scavenging Efficiency of the Two-Stroke Engine: A Comparison and Analysis of Methods - Irish, Karl, Keller, Waldron. - M.I.T. Thesis

FIGURES AND CURVES

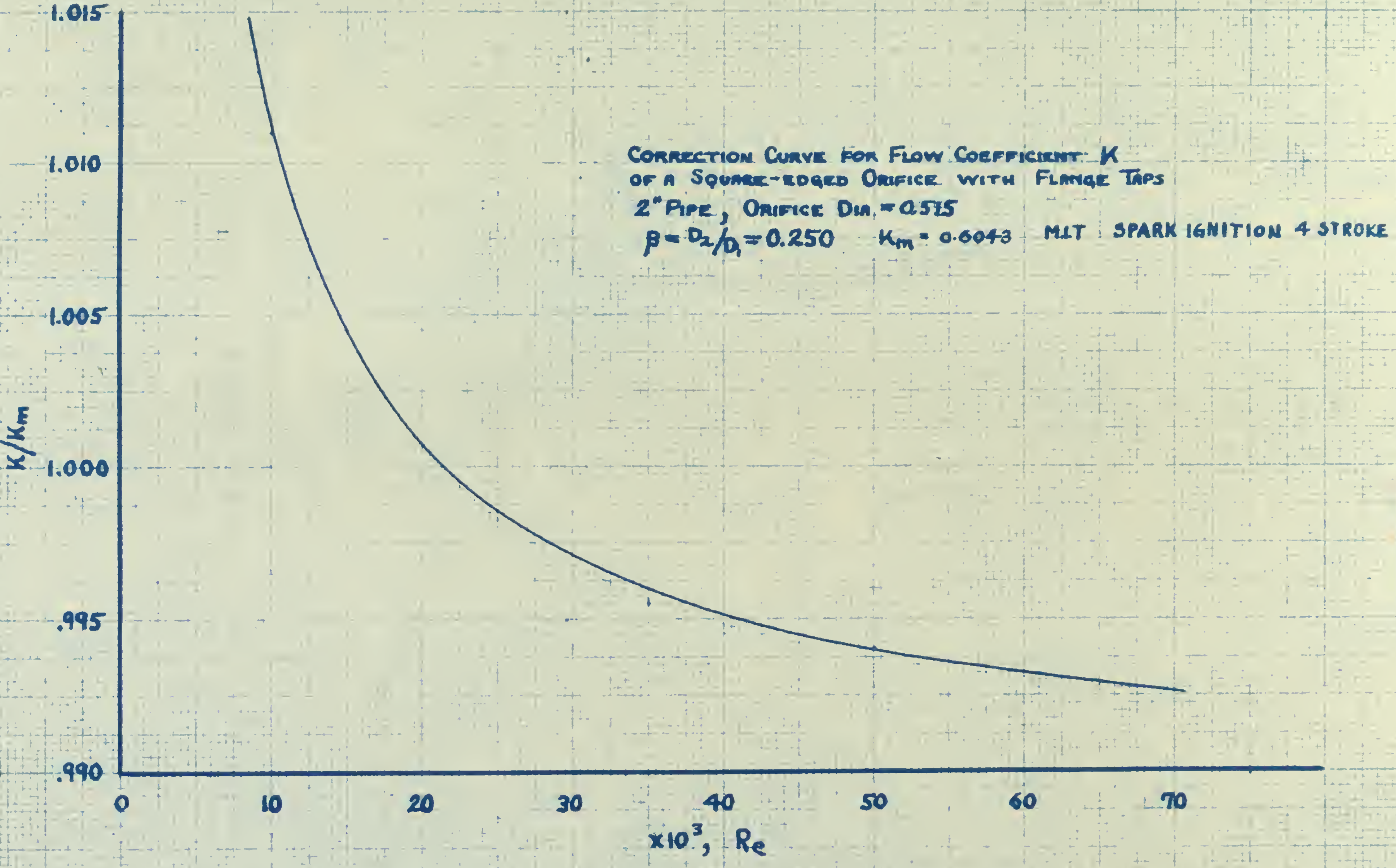
1. Analysis of Exhaust Gases from a Spark Ignition Engine - D'Alleva and Lovell - "Relation of Exhaust Gas Composition to Air-Fuel Ratio" - SAE Journal, Vol. 38, No. 3, March 1936.

2. Relation of Constituents in Normal Exhaust Gas to Fuel-Air Ratio - Gerrish and Meems - Figure 3 of NACA Wartime Report, October 1943. See NACA Report #757.

3. Conditions at End of Expansion, Constant Volume Fuel-Air Cycle - Goodenough and Baker - "A Thermodynamic Analysis of Internal Combustion Engine Cycles" - Univ. of Illinois Eng. Exp. Station, Bulletin 160, 1927.

APPENDIX H

FIGURES



CORRECTION CURVE FOR FLOW COEFFICIENT K
OF A SQUARE-EDGED ORIFICE WITH FLANGE TAPS
2" PIPE, ORIFICE DIA. = 0.575
$\beta = D_2/D_1 = 0.250$ $K_m = 0.6043$ M.I.T SPARK IGNITION 4 STROKE

FIGURE 1

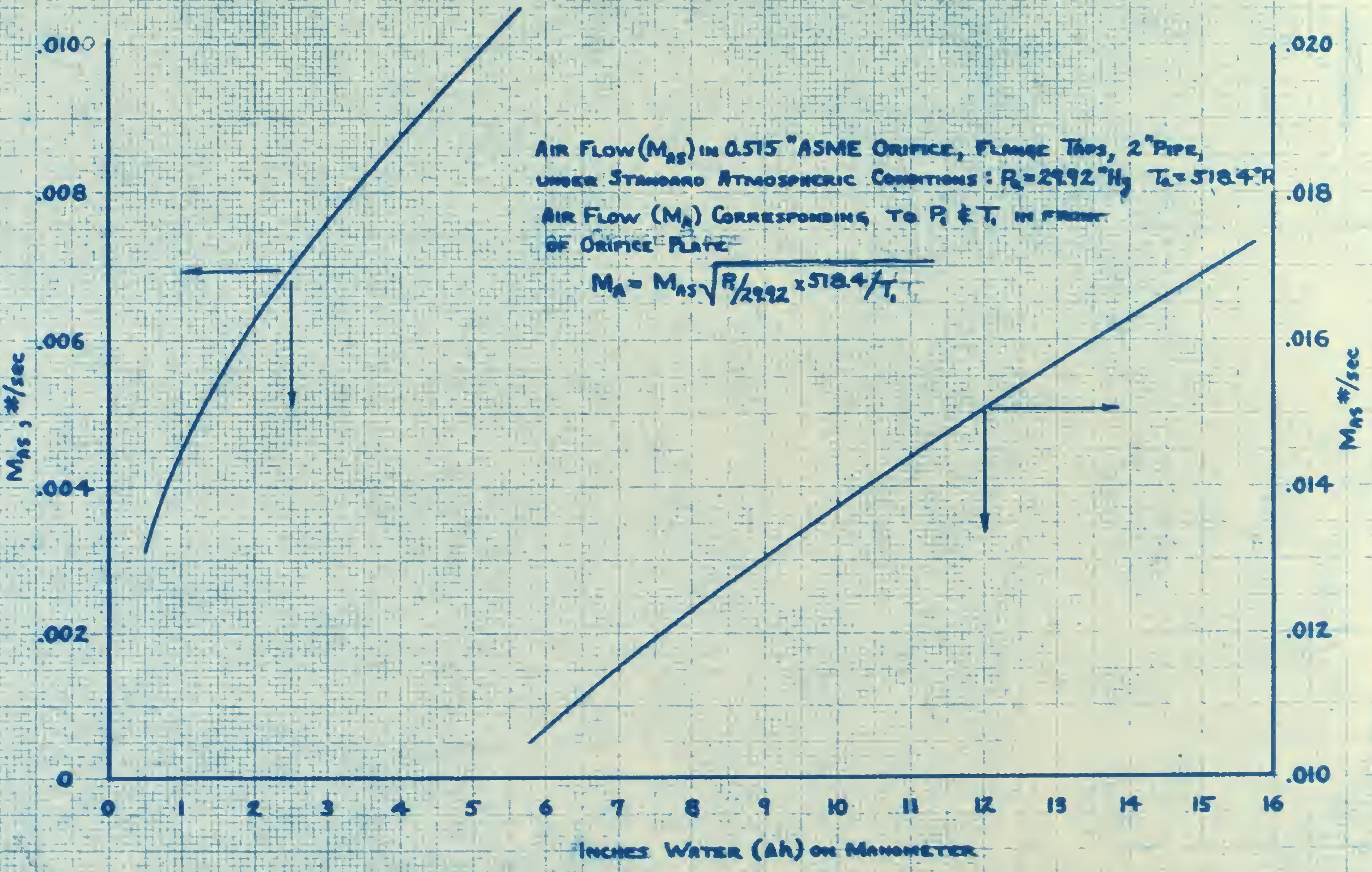
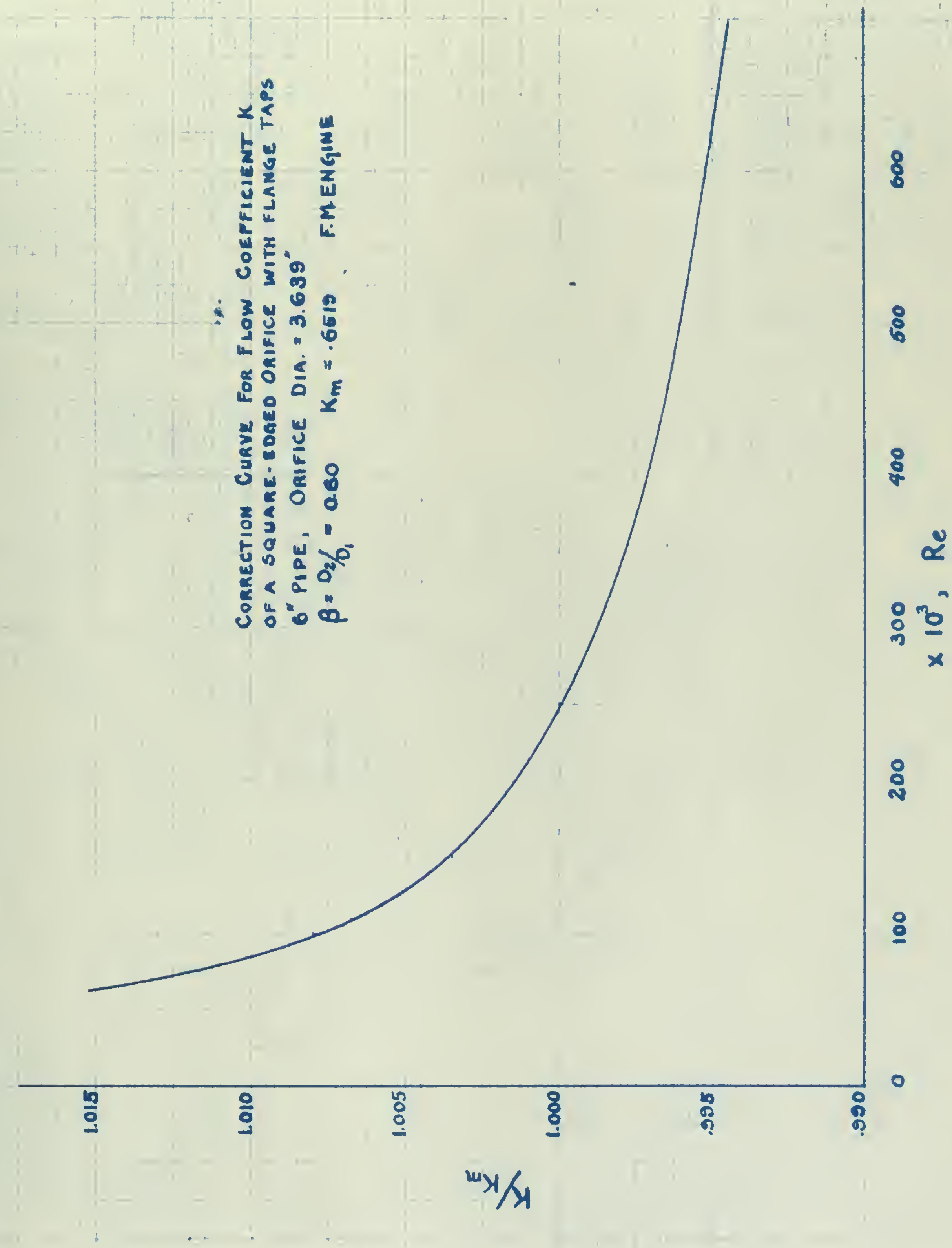
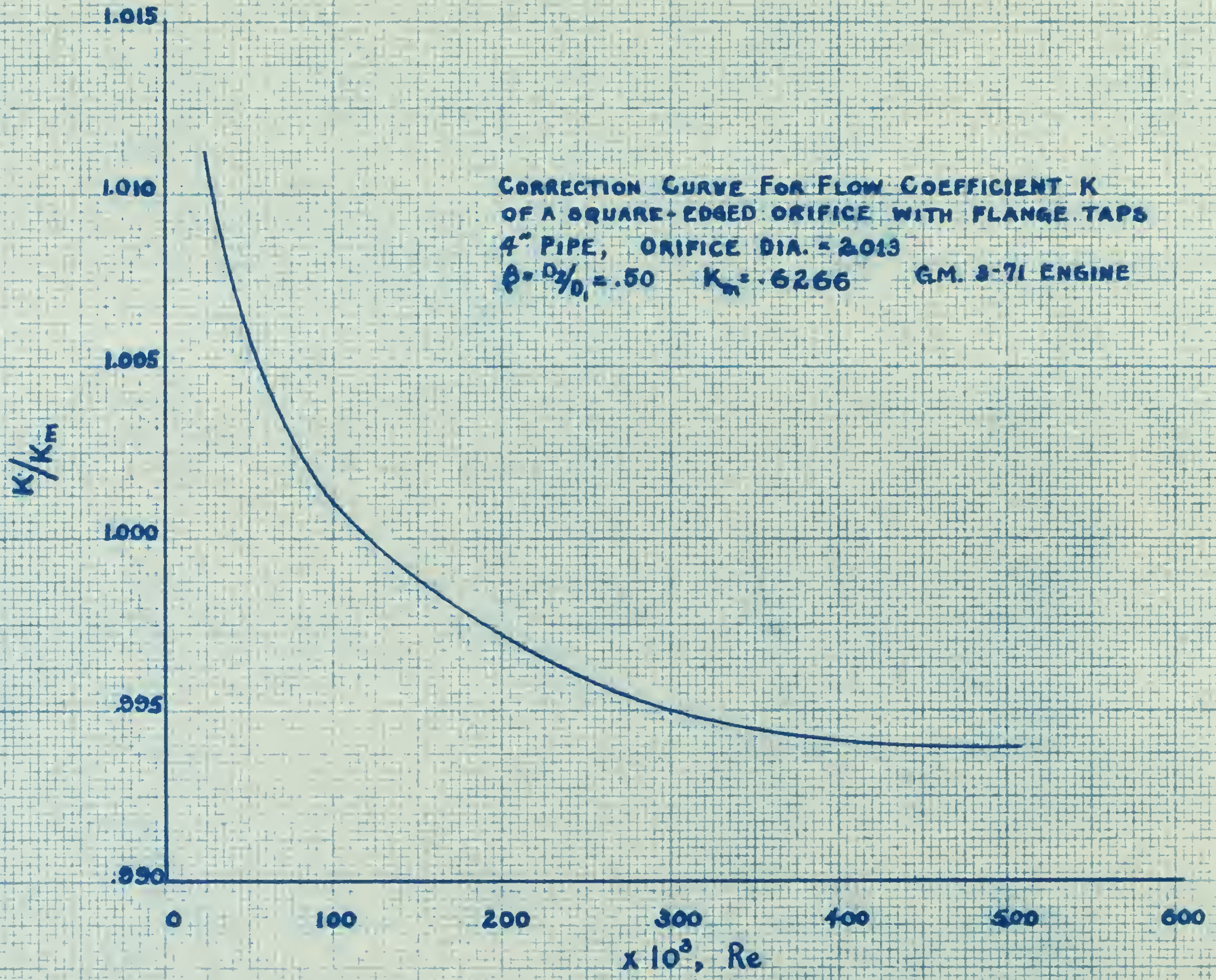

AIR FLOW (M_{AS}) IN 0.515" ASME ORIFICE, FLANGE TAPS, 2" PIPE,
UNDER STANDARD ATMOSPHERIC CONDITIONS: P_a = 29.92" Hg T_a = 518.4°R
AIR FLOW (M_A) CORRESPONDING TO P_1 & T_1 IN FRONT OF ORIFICE PLATE
$M_A = M_{AS}\sqrt{P_1/29.92 \times 518.4/T_1}$
M_{AS}, #/sec
.010
.008
.006
.004
.002
0
M_{AS} #/sec
.020
.018
.016
.014
.012
.010
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
INCHES WATER (Δh) ON MANOMETER

FIGURE 2

FIGURE 3

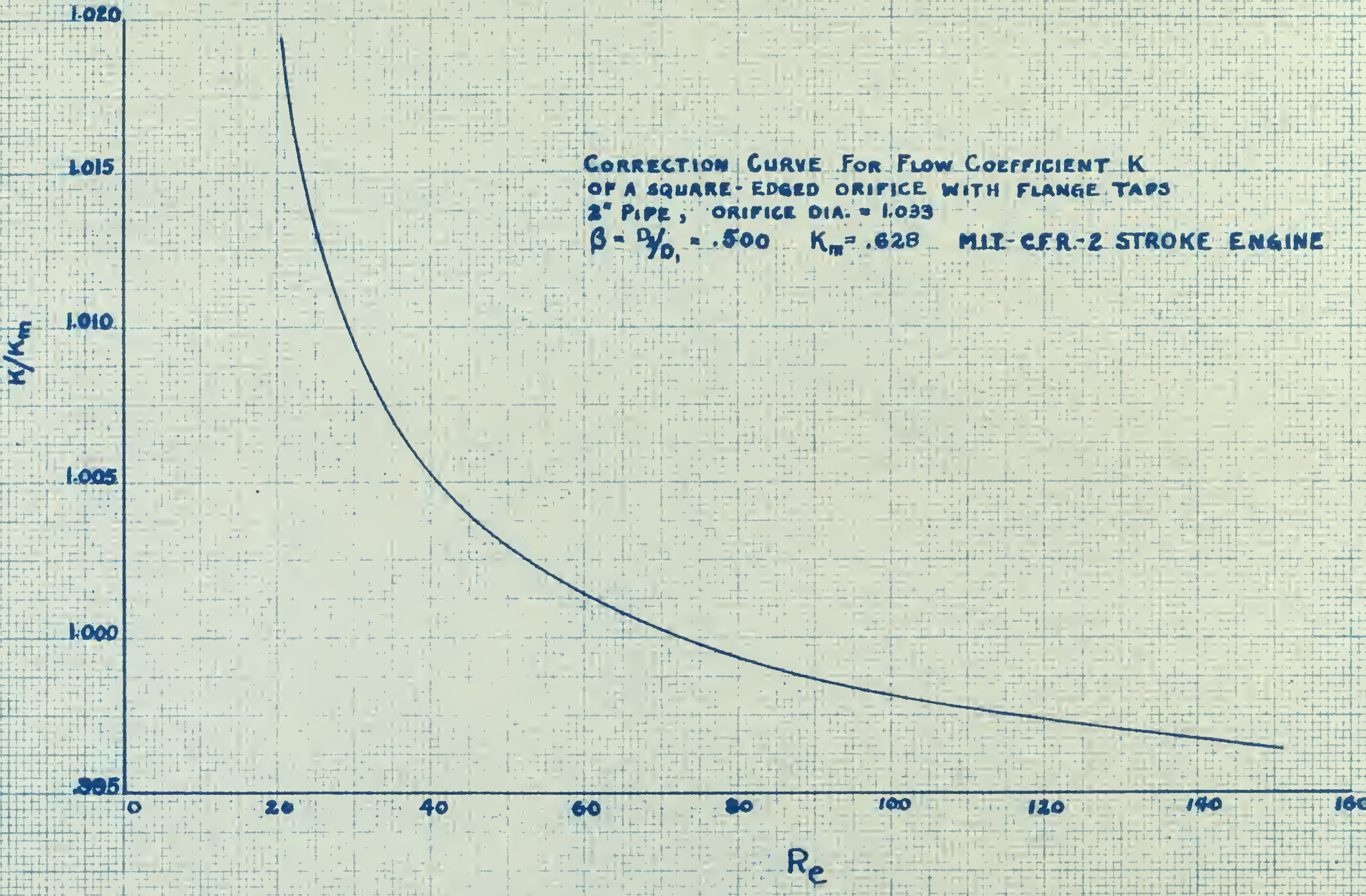


CORRECTION CURVE FOR FLOW COEFFICIENT K
OF A SQUARE-EDGED ORIFICE WITH FLANGE TAPS
6" PIPE, ORIFICE DIA. = 3.639"
$\beta = D_2/D_1 = 0.60$ $K_m = .6519$ F.M. ENGINE



CORRECTION CURVE FOR FLOW COEFFICIENT K
OF A SQUARE-EDGED ORIFICE WITH FLANGE TAPS
4" PIPE, ORIFICE DIA. = 2.013
$\beta = D_2/D_1 = .50$ $K_m = .6266$ G.M. 3-71 ENGINE

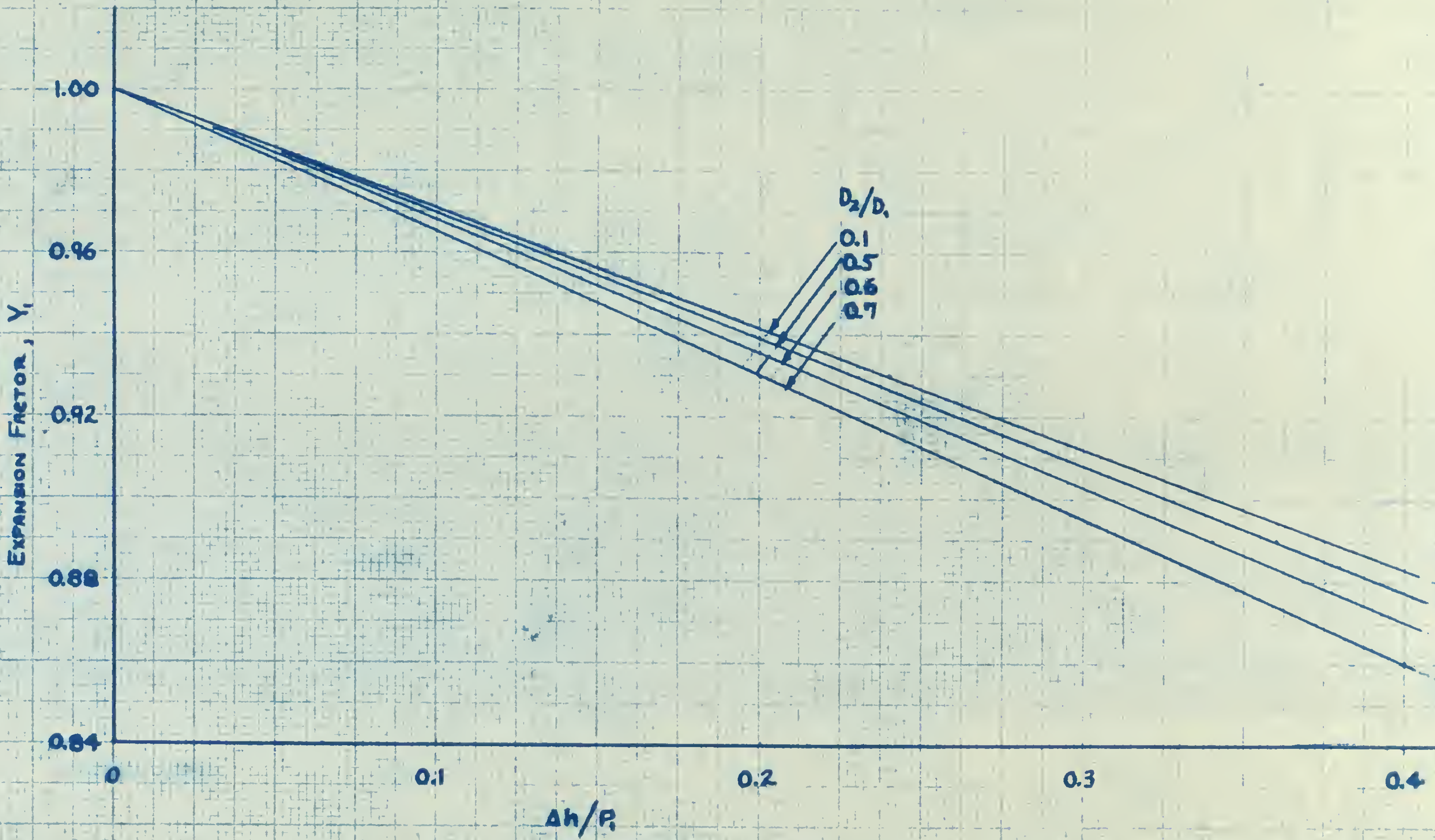
FIGURE 4



CORRECTION CURVE FOR FLOW COEFFICIENT K
OF A SQUARE-EDGED ORIFICE WITH FLANGE TAPS
2" PIPE, ORIFICE DIA. = 1.033
$\beta = D/D_1 = .500$ $K_m = .628$ M.I.T.-C.F.R.-2 STROKE ENGINE

FIGURE 5

FIGURE 6



EXPANSION FACTORS FOR AIR

$k = 1.4$; COMPUTED FROM $Y_1 = 1 - [0.41 + 0.35(D_2/D_1)^4][\Delta h/P_1/k]$

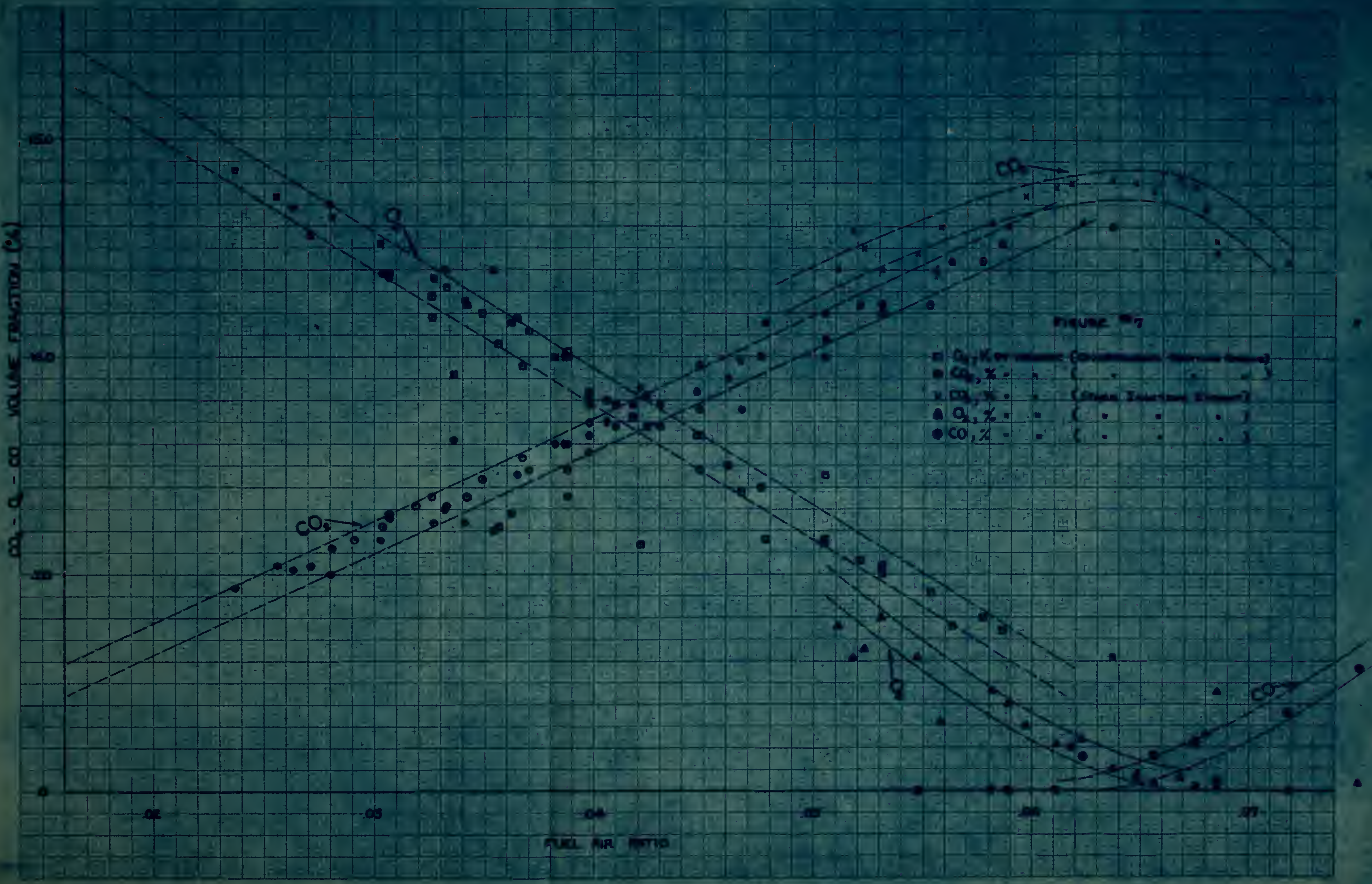
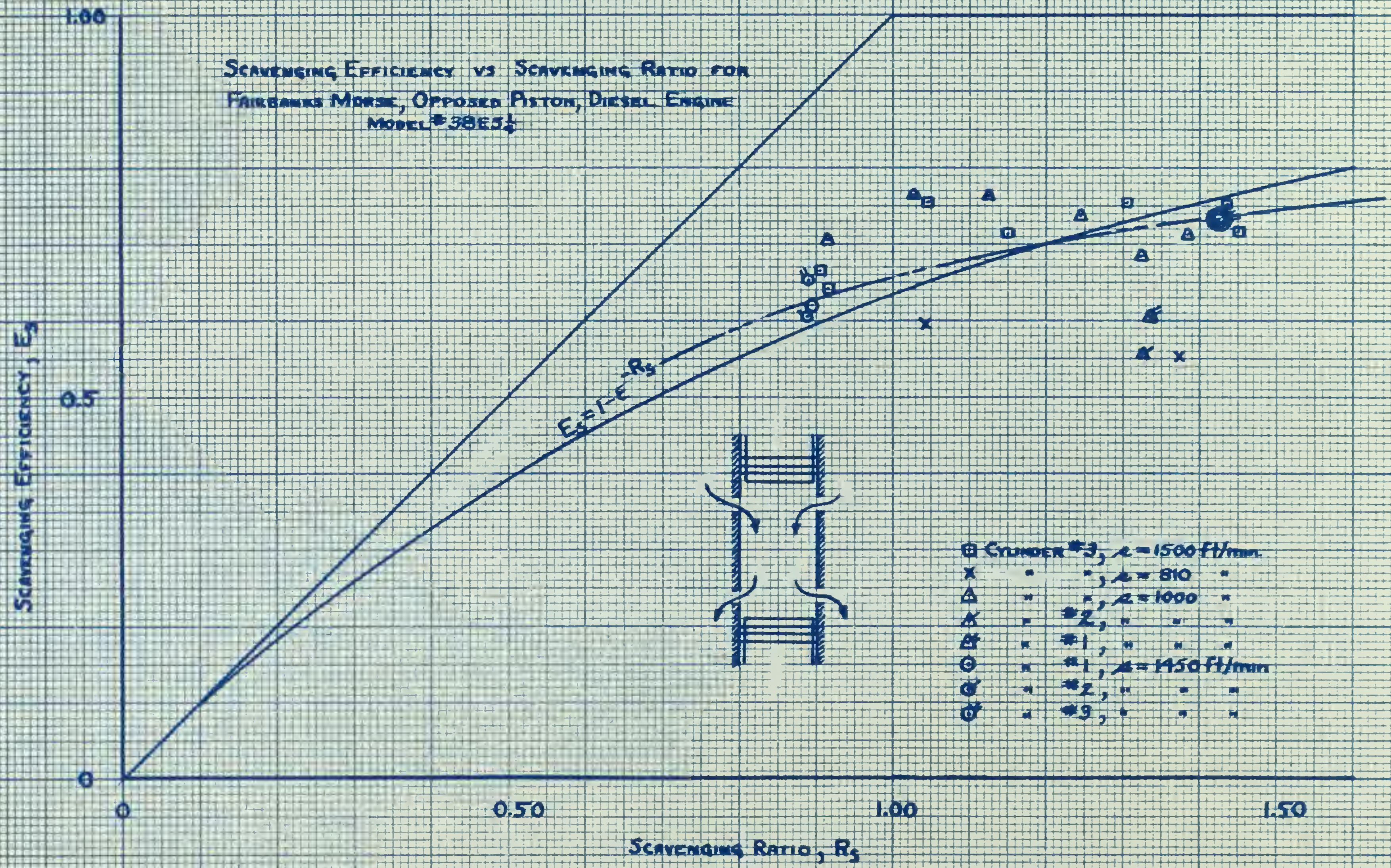
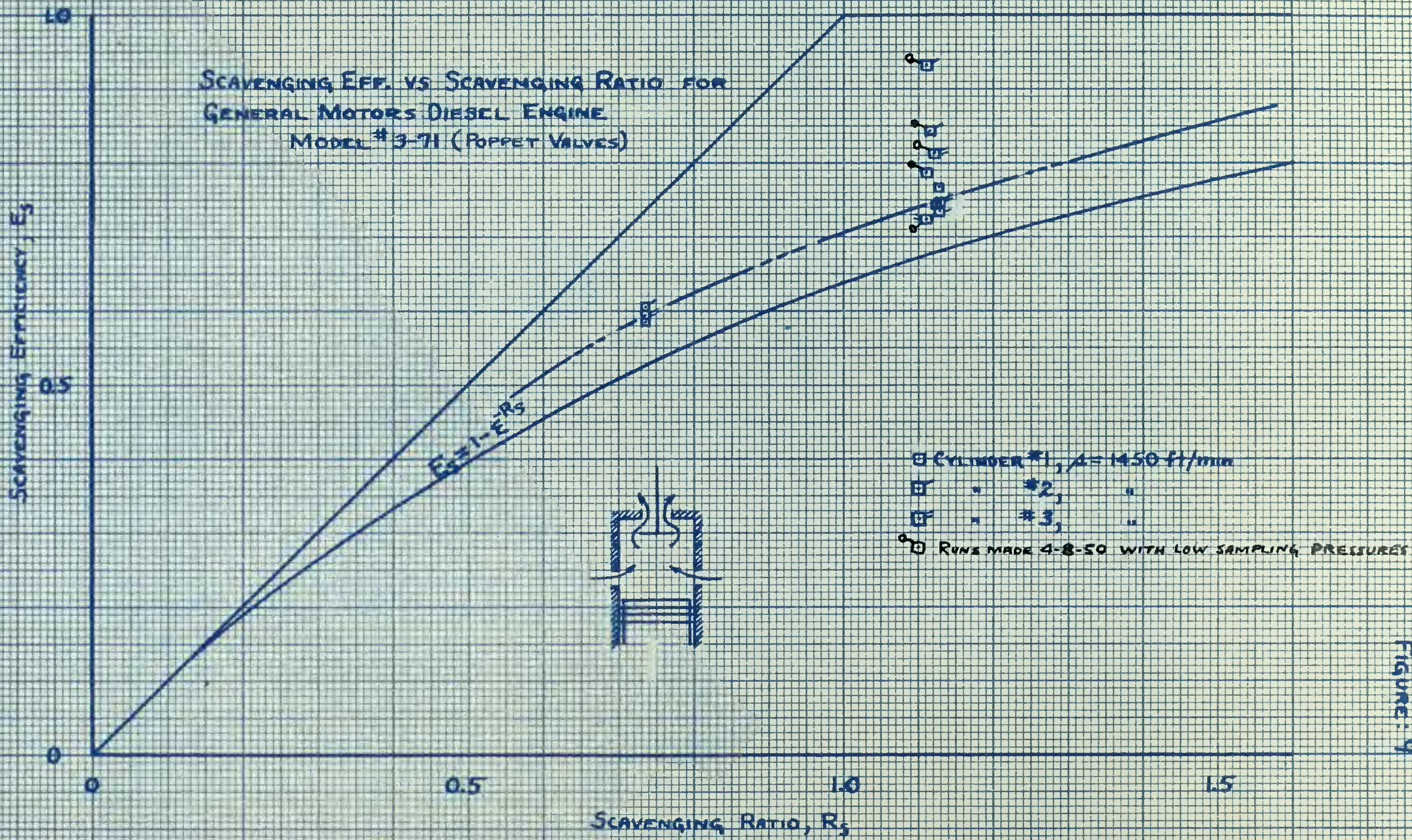


FIGURE #7
O₂, % by volume (Continuous [illegible])
CO₂, % " " (" " ")
CO₂, % " " (Stein Injection [illegible])
O₂, % " " (" " ")
CO, % " " (" " ")
CO₂
O₂
CO₂
O₂
CO₂
CO
CO₂ - O₂ - CO VOLUME FRACTION (%)
15.0
10.0
5.0
0
.02
.03
.04
.05
.06
.07
FUEL AIR RATIO



SCAVENGING EFFICIENCY VS SCAVENGING RATIO FOR
FAIRBANKS MORSE, OPPOSED PISTON, DIESEL ENGINE
MODEL #38E5¼

FIGURE: 8



SCAVENGING EFF. VS SCAVENGING RATIO FOR GENERAL MOTORS DIESEL ENGINE MODEL #3-71 (POPPET VALVES)

FIGURE: 9

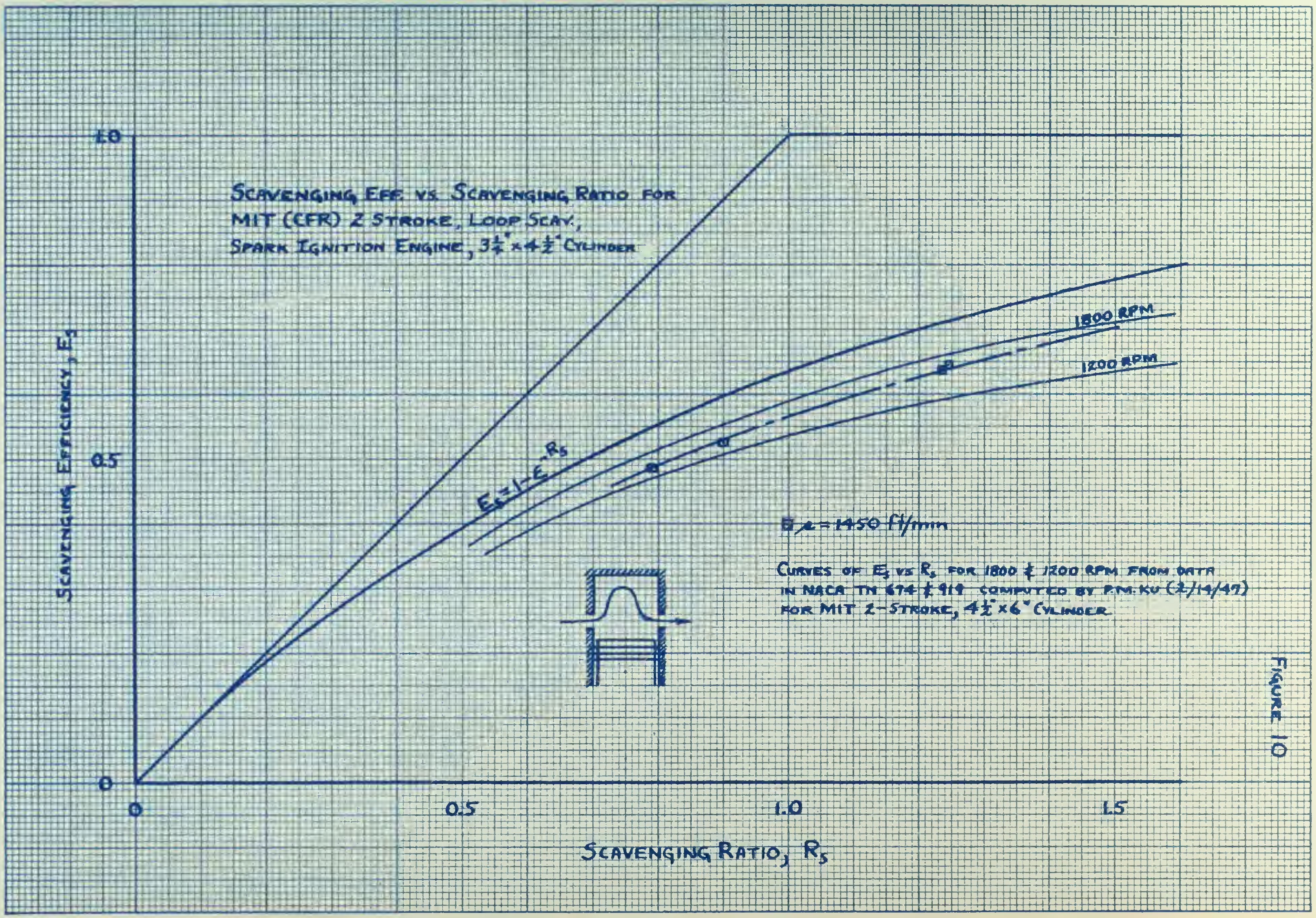
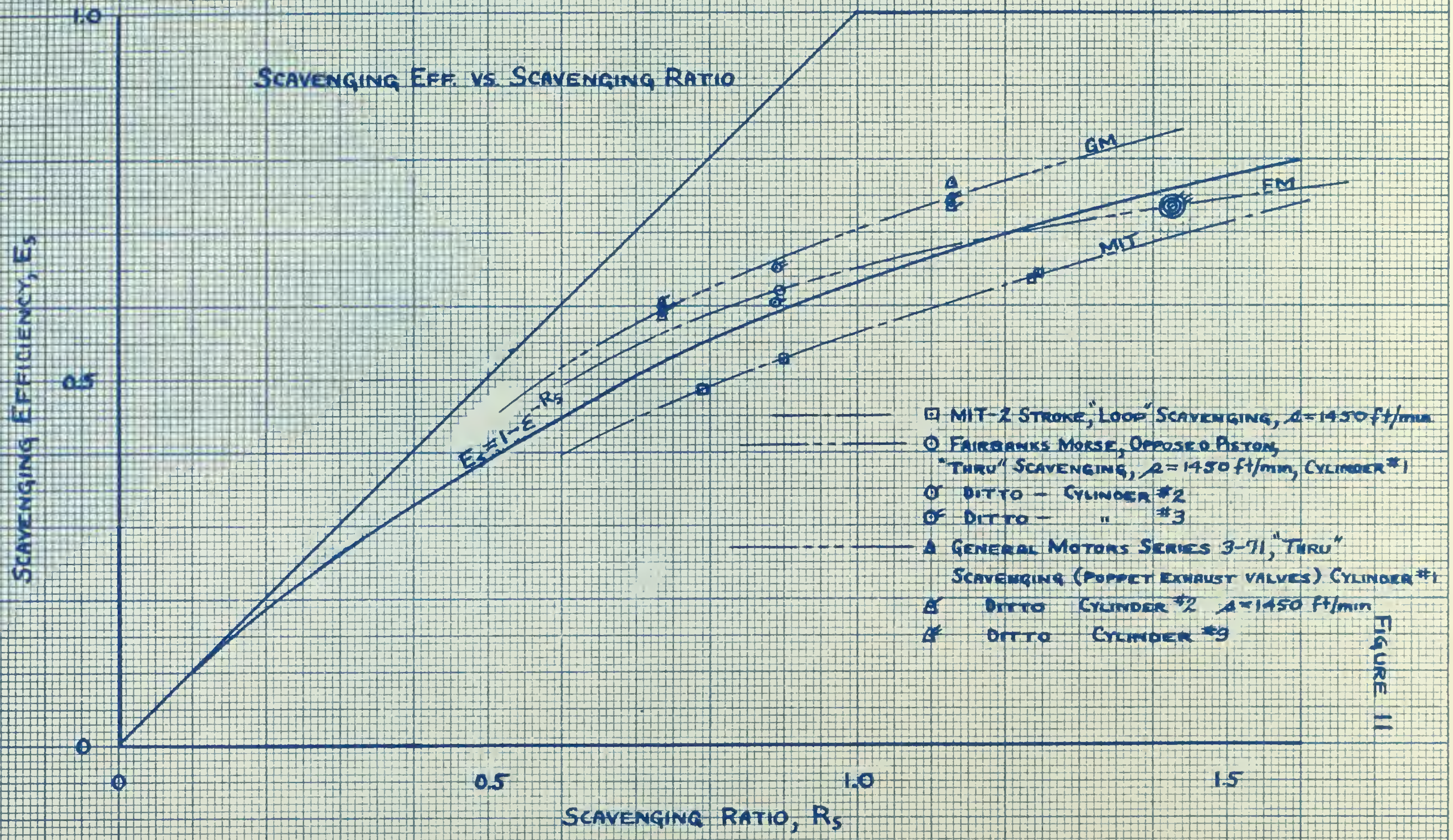

Scavenging Eff. vs. Scavenging Ratio for
MIT (CFR) 2 Stroke, Loop Scav.,
Spark Ignition Engine, $3\frac{1}{4}'' \times 4\frac{1}{2}''$ Cylinder
$E_s = 1 - \varepsilon^{-R_s}$
1800 RPM
1200 RPM
$\bar{S}_p = 1450$ ft/min
Curves of E_s vs R_s for 1800 & 1200 RPM from data
in NACA TN 674 & 919 computed by P.M. Ku (2/14/47)
for MIT 2-Stroke, $4\frac{1}{2}'' \times 6''$ Cylinder
Scavenging Efficiency, E_s
Scavenging Ratio, R_s
1.0
0.5
0
0
0.5
1.0
1.5

FIGURE 10



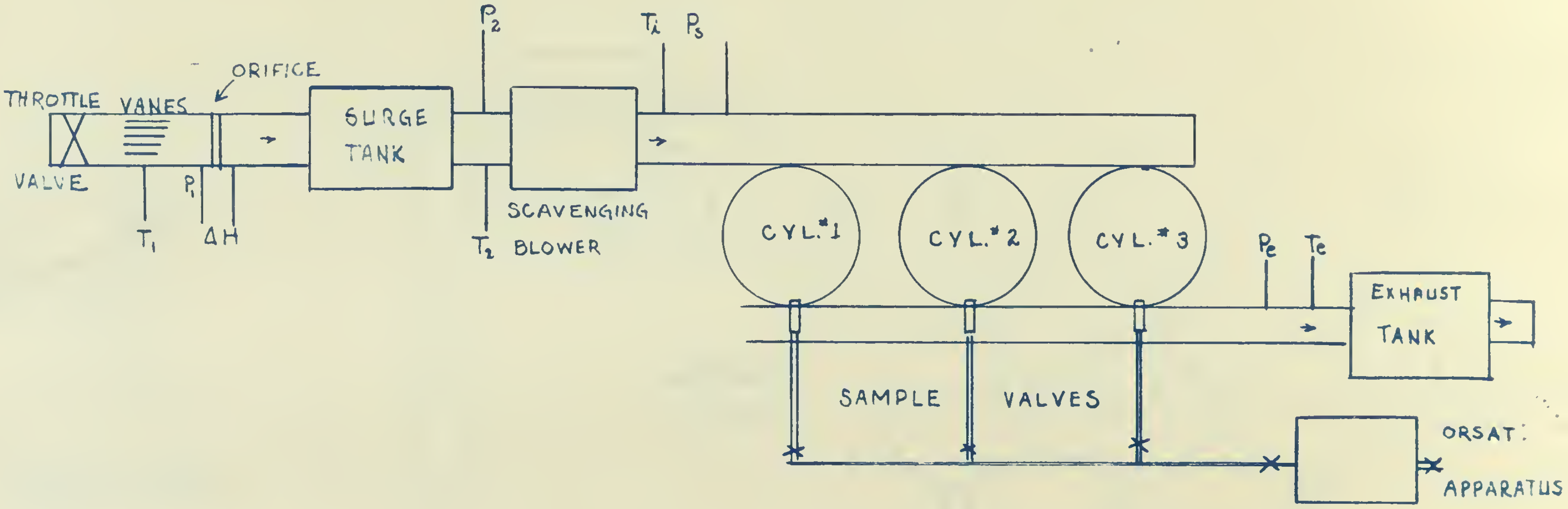
SCAVENGING EFF. VS. SCAVENGING RATIO

FIGURE 11

APPENDIX I

PLATES

PLATE I



SCHEMATIC DIAGRAM OF G.M.
SERIES 3-71 ENGINE SET-UP

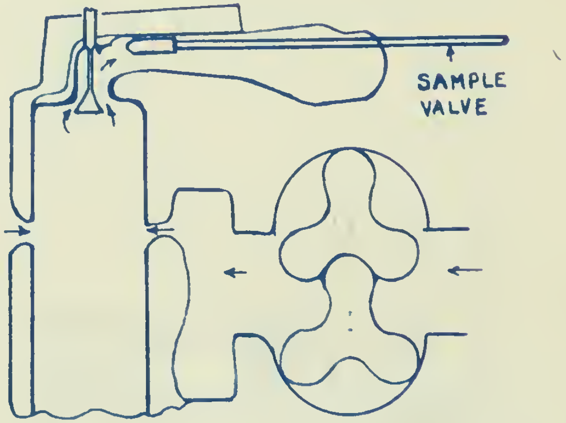


DIAGRAM OF AIR FLOW SYSTEM

PLATE II

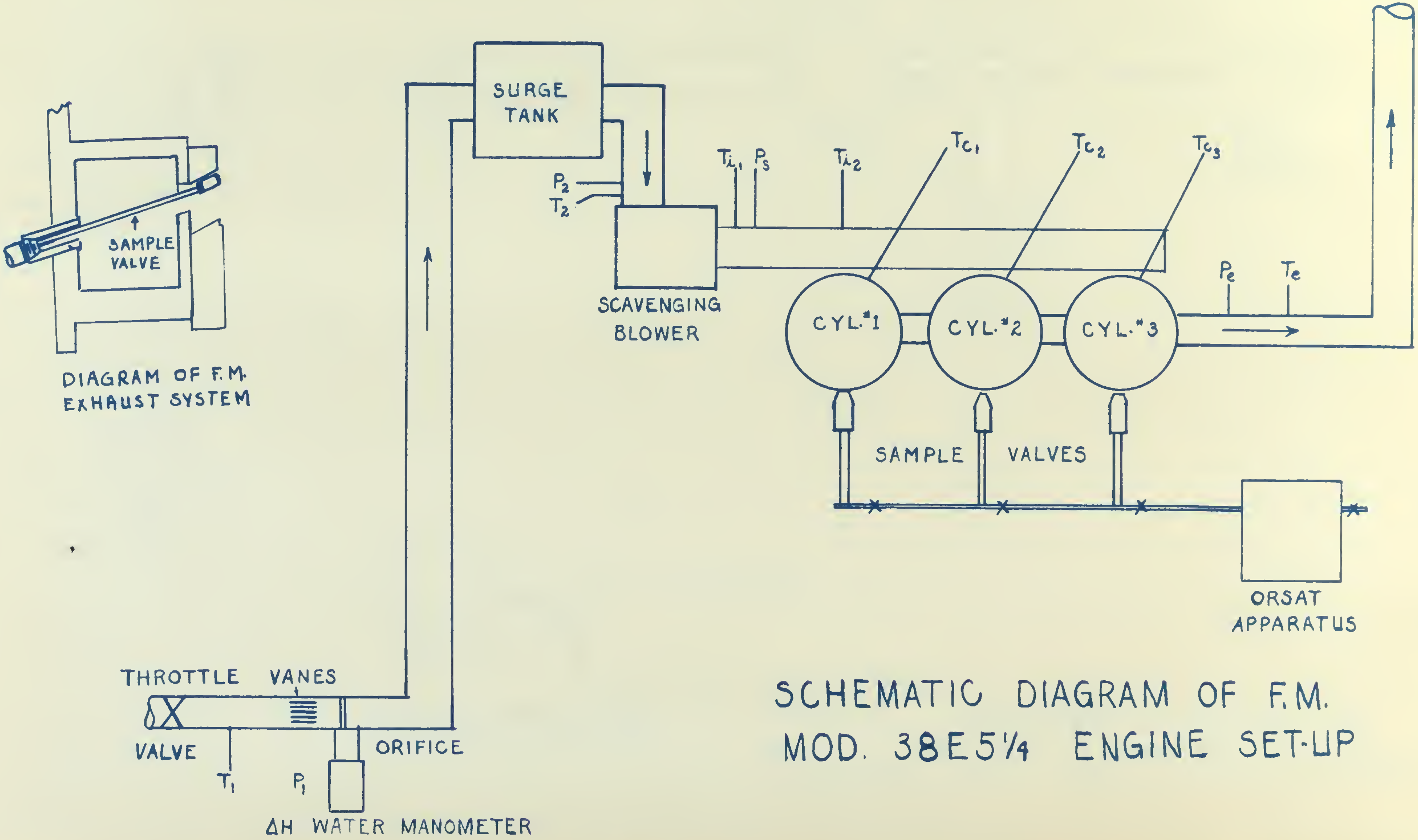
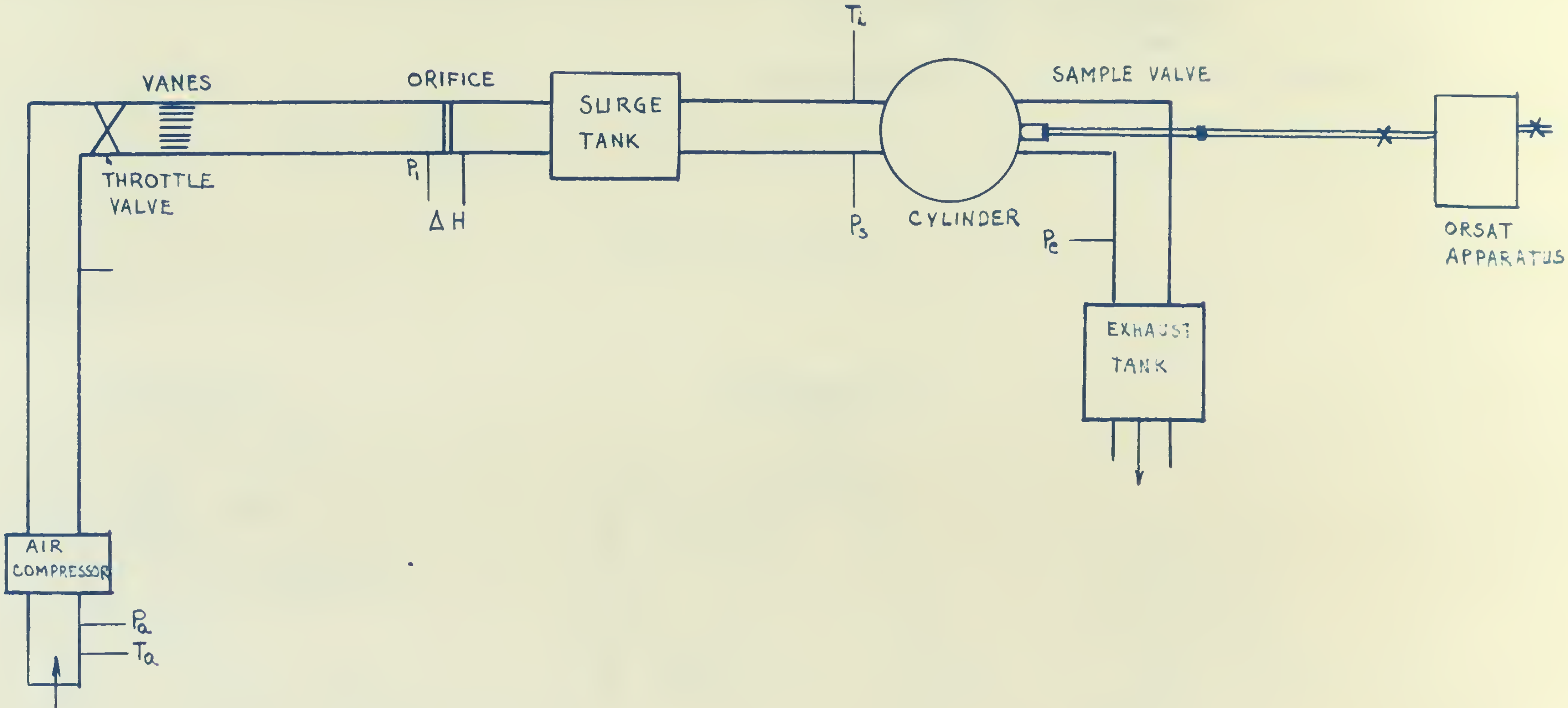
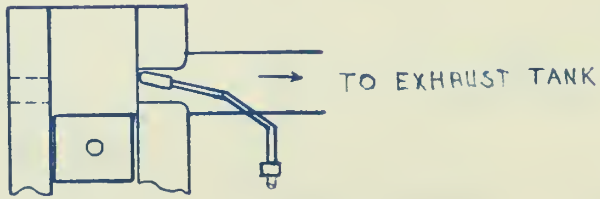


DIAGRAM OF F.M. EXHAUST SYSTEM

SCHEMATIC DIAGRAM OF F.M. MOD. 38E5¼ ENGINE SET-UP

PLATE III

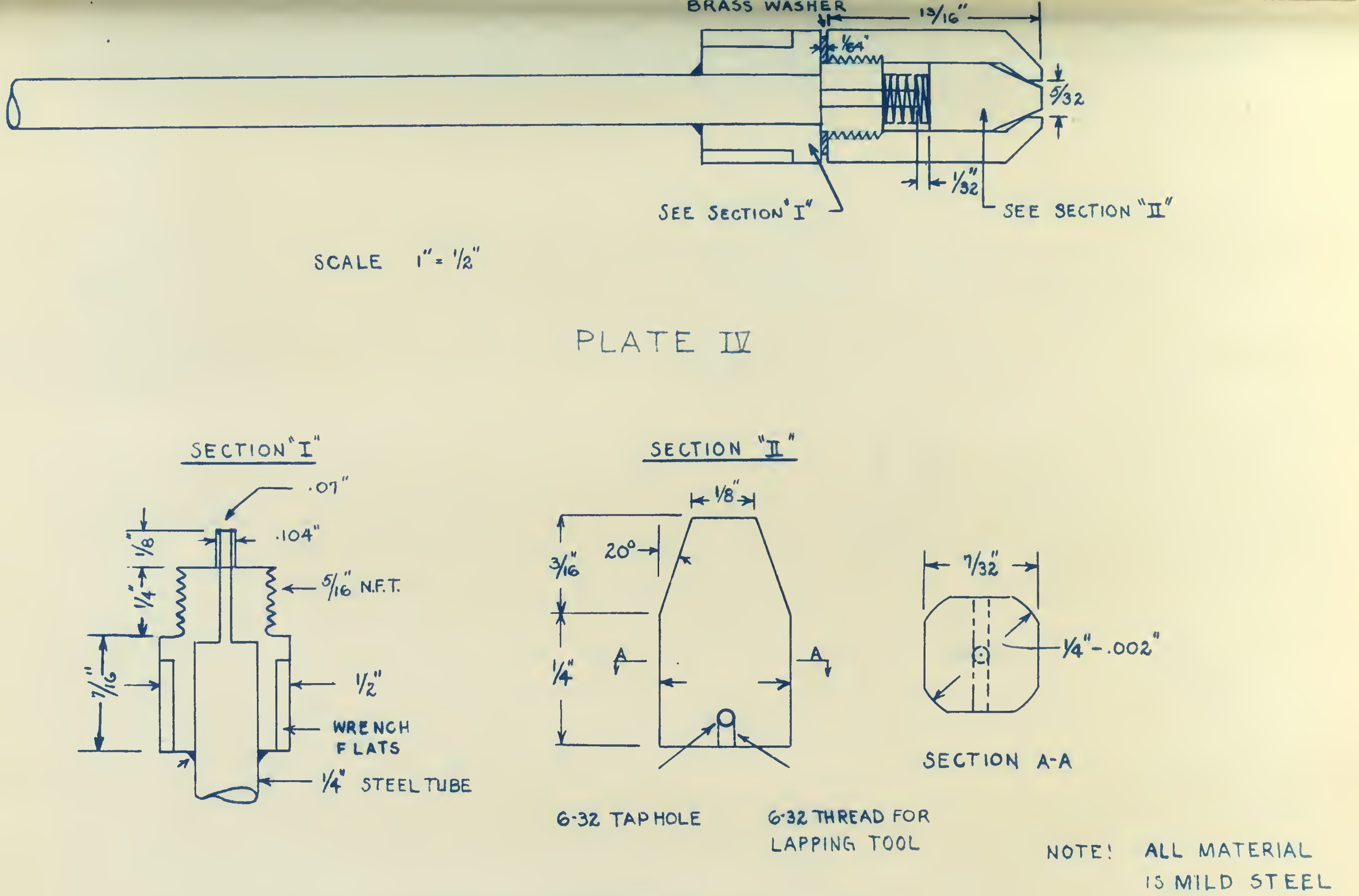




CROSS SECTION OF CYLINDER
SHOWING LOCATION OF THE
SAMPLING VALVE

SCHEMATIC DIAGRAM OF
M.I.T. C.F.R. 2 STROKE ENGINE

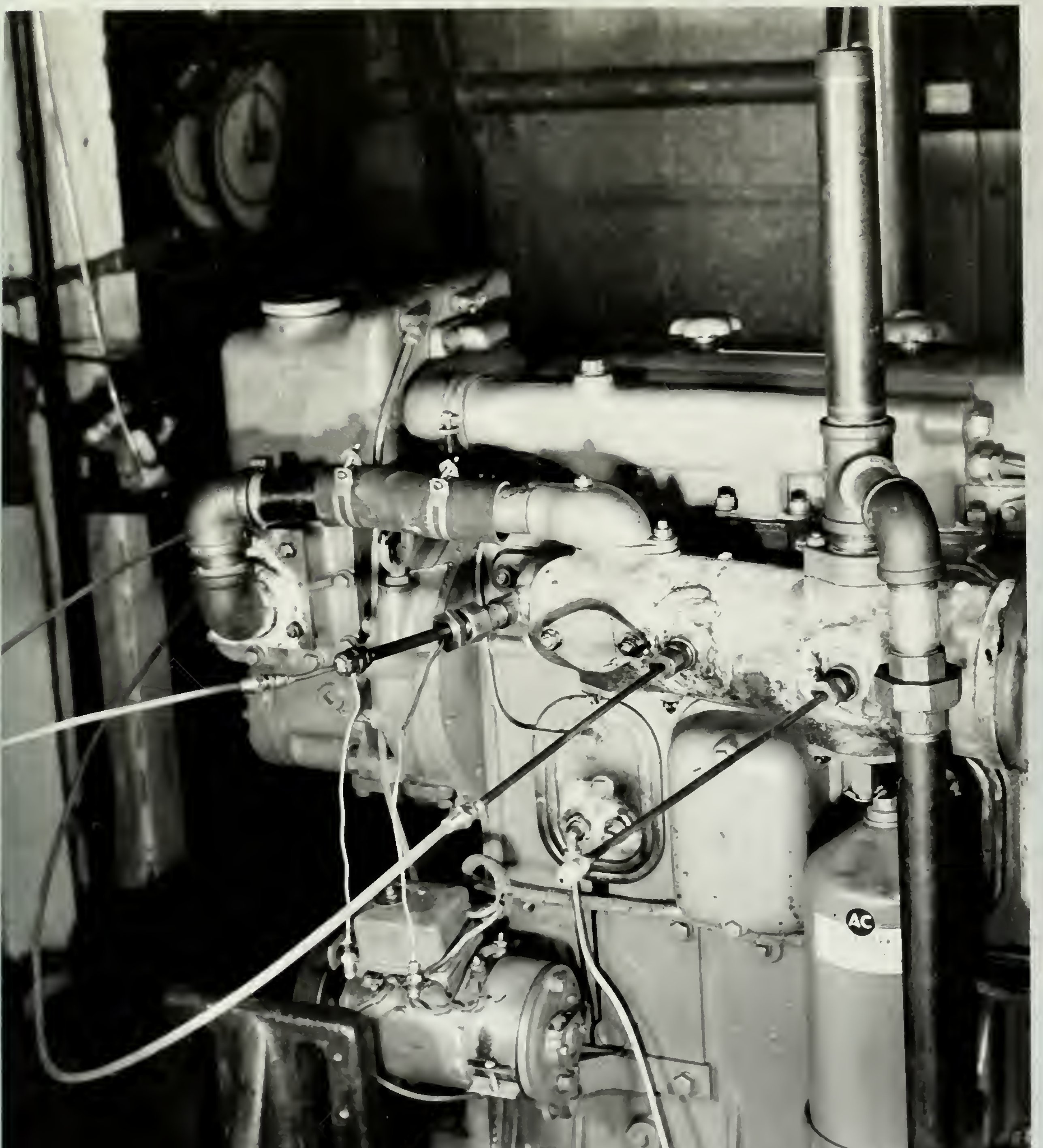
PLATE IV



DETAILS OF SAMPLING VALVE

APPENDIX J

PHOTOGRAPHS



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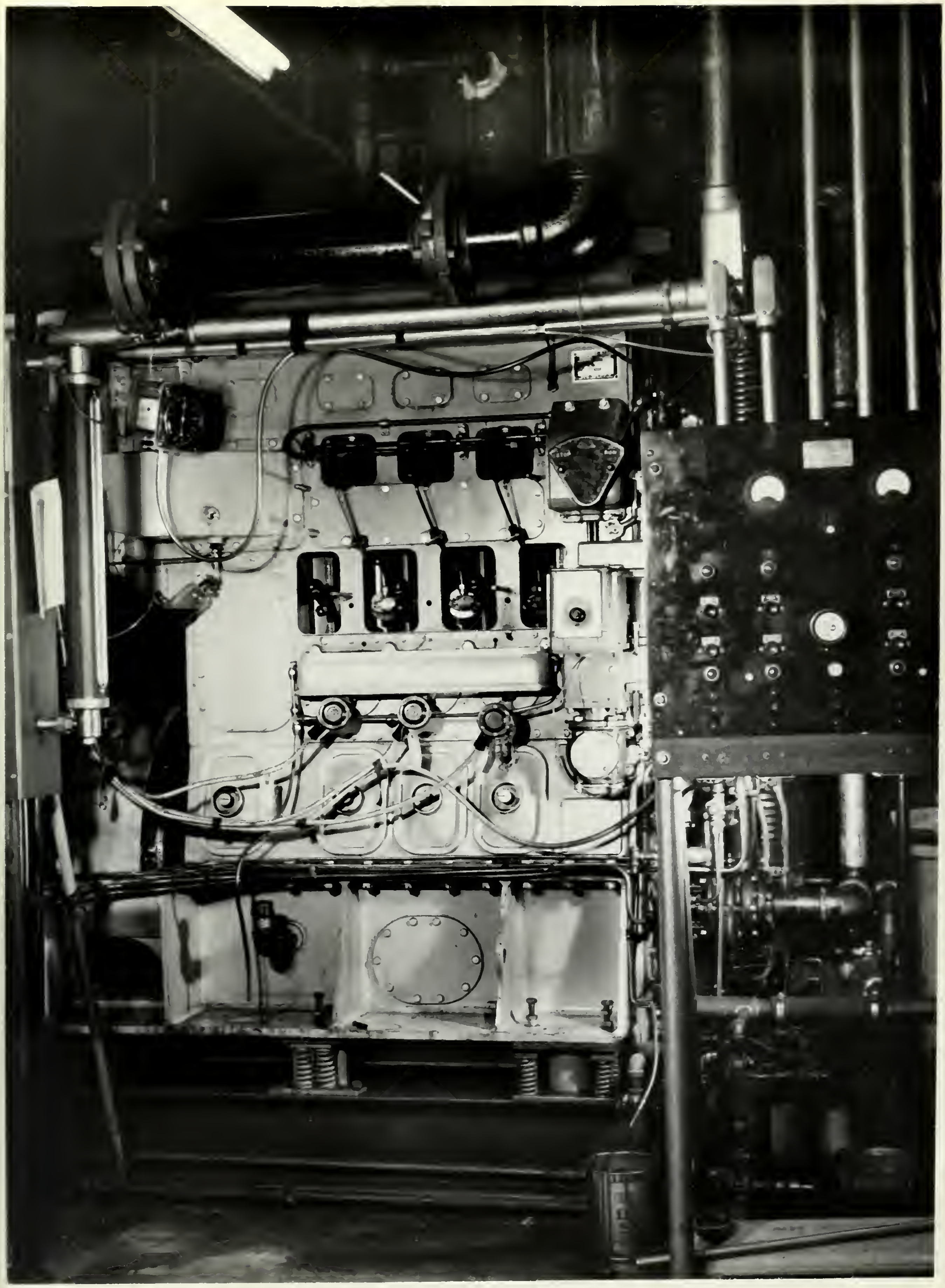

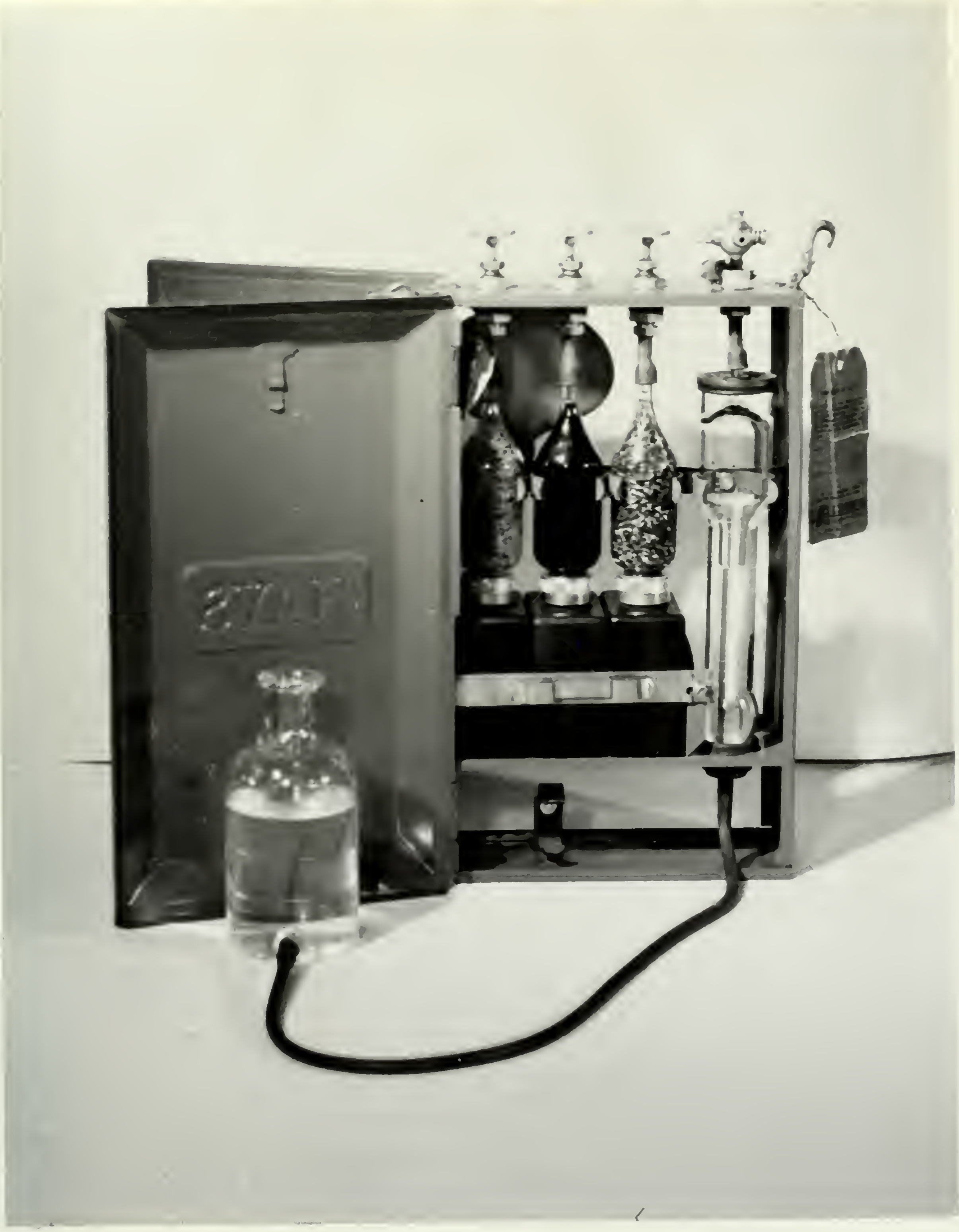

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